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U. S. S. *MARYLAND.*

DESCRIPTION AND OFFICIAL TRIALS.

BY LIEUTENANT COMMANDER F. T. VAN AUKEN, U. S. N., MEMBER.

The U. S. S. *Maryland* was authorized by Act of Congress 29 August, 1916, at a contract price, to include hull and machinery, of cost plus fee. The other ships of the *Maryland* class authorized under this Act were the *West Virginia, Colorado* and *Washington.* These ships, together with the *Tennessee* and *California,* whose major difference from the *Maryland* class is in battery alone, were built as a result of the study of the last war, particularly with respect to water tight integrity, arrangement and design of battery and the use of turbo generators and motors for main propulsion.

The contract for the ship was signed with the Newport News Shipbuilding and Dry Dock Co. on 5 Dec., 1916, with supplementary contracts dated 29 May, 1917, and 18 June, 1920. The General Electric Co., of Schenectady, N. Y., was the sub-contractor for main propelling machinery, the six 300 K.W. turbo generating sets and the necessary spare parts.

The keel was laid 24 April, 1917; the ship was launched 20 March, 1920; preliminary trials by the contractor were held off the Virginia Capes on 20 June, 1921, and the ship commissioned 21 July, 1921, at the Navy Yard, Norfolk, Va. The official machinery and gunnery trials, resulting most satisfactorily, were held November 14-17, 1921, off Rockland, Maine, and en route Rockland to New York. Subsequent to this last date the *Maryland* has had five months active cruising experience in which she has fully demonstrated both her reliability and efficiency. An example of this was an 18.5 knot continuous run from Rio de Janeiro, Brazil, to New York, extending over a period of ten days and sixteen hours in which no speed change was made or was necessary (the ship ran through an 85-mile storm), no casualties to machinery occurred and the run concluded with a four-hour high power trial at 21.0 knots. Such a run through the length of the tropics at high temperatures is unquestionably the severest test to which electric propulsion machinery can be subjected and the performance of the *Maryland's* equipment was most satisfactory.

HULL DATA.

Length between perpendiculars, feet	600
Projection forward of for, perpendicular, feet	24
Length overall, feet	624
Length of straight keel, feet	539
Breadth, moulded, feet	96
Draft, L.W.L., designed above bottom of keel, feet and inches	30–6
Displacement, normal at 30 ft. 6 in. draft, tons	32,630

At designed draft	Area immersed midship section, sq. ft	2,904
	Area L.W.L. plane, sq. ft	42,430
	Wetted surface, sq. ft	76,611
Coefficient of fineness, block		.634
	midship section	.977
	L.W.L.	.725
Fuel oil, 95 per cent capacity of compartments, tons		5,000
Reserve feed water, capacity of compartments, tons		575
Fresh water tanks, capacity, tons		175

NOTE: The displacement includes two-thirds full supply of stores, fuel and water; full supply of ammunition.

GENERAL DESCRIPTION OF HULL.

The *Maryland* and *Tennessee* class of ships markedly differ in profile from any other capital class ship in the United States Navy, the outstanding difference being the very large fire control tops fitted on the two conventional cage masts. These tops, together with the new large standard bridge structures, throw the two smoke stacks in marked relief, causing the latter to appear altogether diminutive in size. The cage masts are provided with the usual wireless and signal yards, the mainmast alone having a topmast. The mainmast is also surrounded with platforms for high power searchlights, these four comprising the ship's entire searchlight equipment, with the exception of the signal searchlights on the bridge.

SUPERSTRUCTURE DECK AND BRIDGES.

The superstructure deck extends from the barbette of turret No. 2 at frame 50 forward to frame 89 aft. On this deck are mounted two 5-inch guns, one on each side at frame 52; four six-pounder saluting guns, two on each side about frame 56; eight 3-inch anti-aircraft guns, four on each side. On this deck are also fitted the garbage incinerator, boat cranes, and boat stowage for the 14 boats, all of which, except the

race boats, are gas-engine driven. The standard bridge structure contains ten staterooms; two baths and two toilets—one each on the two upper levels; ventilating trunks for the forward engineroom and the four forward firerooms; the signal bridge; charthouse; flag radio room; navigating bridge and glass inclosed pilot house.

UPPER DECK.

This deck extends from the bow to frame 89 and is a weather deck to frame 50 and, between frames 50 and 89, is covered by the superstructure deck—there being a weather deck passageway outboard on each side along its whole length. The windlass, anchor chains, two electric deck winches and turrets No. 1 and No. 2 are forward, while in the compartments covered by the superstructure are ten of the twelve 5-inch guns in casemates; the crew's and officer's galleys; combined blacksmith, coppersmith, pipe and foundry shops; the bakery and butcher shop.

MAIN DECK.

The main deck is a weather deck aft of the break in the upper deck at frame 89. On this deck are mounted turrets No. 3 and No. 4; two electric deck winches; electric capstan; pneumatic catapult for launching airplanes and supply and exhaust trunks for ventilation to main propulsion spaces. The covered portion forward contains crew's living quarters, reception room and library; laundry and ship's store; the following offices: Chaplain, Flag, deck and Dental; carpenter and barber shops; the main commissary store and issuing room; general mess pantry; chief petty officers' messroom and pantry; the following wash rooms: mess attendants, crew's and chief petty officers, together with a stowage space for running gear and tackle and sand locker.

SECOND DECK.

On this deck are crew's, chief petty officers and officers living quarters; sick bay, dispensary, and the following offices: Communication, Decoding, Flag, Gunnery, Captain's, Engineer's, First Lieutenant, Pay and Executive.

THIRD DECK.

On this deck are located crew's and officer's living quarters, store rooms, windlass machinery, electric driven air compressors, submersible pump equipment with motor generators, battle dressing station, general work shop with very complete machine tool equipment, repair shops for sundry electric machinery, main radio rooms, kite balloon winch, boat crane machinery, capstan machinery, cold storage room, armory and fuel oil relay tanks and control valve equipment for handling all fuel oil for delivery and use.

FIRST PLATFORM, SECOND PLATFORM AND HOLD.

On these decks are located machinery spaces, magazine and shell stowage, turret handling rooms, torpedo room, fuel oil stowage tanks, reserve feed bottoms, battery control rooms and various storerooms.

COMPLEMENT.

Commanding Officer	1
Ward Room Officers	29
Junior Officers	27
Warrant Officers	9
Seaman branch	739
Artificer branch	121
Engine room force	263
Commissary branch	24
Special branch, including yeoman, storekeepers, pharmacists, hospital apprentices, buglers and musicians	78
Messman branch	41
Total	1,332

BATTERY.

The main battery consists of eight 16-inch 45 caliber guns mounted in four turrets of two guns each, two turrets forward and two aft, all on the center line. The guns are mounted in individual slides and can be elevated independently.

The secondary battery consists of twelve 5-inch 51 caliber rapid fire guns, two of which are mounted on the superstructure deck and the other ten—five on each side—in the inclosed portion of the upper deck. The 5-inch battery is served by eighteen 3 H.P. electric driven chain ammunition hoists, variable speed in hoisting only, designed to deliver horizontally a maximum of twenty pieces of ammunition per hoist per minute, the minimum rate of delivery being 13.3 pieces of ammunition per hoist per minute. In addition to the hoists there are also installed four horizontal continuous motion reversible conveyors for transfer of powder cases along the splinter deck passages to the port and starboard amidship hoists.

Additional guns consist of eight 3-inch anti-aircraft and four six pounders for saluting—all mounted on the superstructure deck; two one-pounder boat guns; one 3-inch field piece and two .30 caliber machine guns.

The torpedo equipment consists of two 21-inch submerged torpedo tubes mounted in the ship's side just above the second platform between frames 15 and 24.

SMALL BOATS.

2—50-foot motor sailers.
2—40-foot motor sailers.
2—26-foot motor life boats.
3—50-foot motor boats.
1—40-foot motor boat.
2—35-foot motor boats.
2—31-foot racing cutters (oars).
2—30-foot whale boats (oars).

All the above boats, with the exception of the whale boats, are stowed on the superstructure deck and all are gas engine drive with the exceptions as noted above. Particular attention is drawn to the fact that no steam boats are provided nor has experience dictated the need of such.

BOAT CRANES.

All boats stowed on the superstructure deck are handled by two pillar type cranes, one on each side of the superstructure deck at frame 74. These cranes are designed for a working load of 40,000 pounds, assuming heel of the vessel 10 degrees in direction of overhang of crane. For handling light loads, a whip is fitted with a speed of eighty feet per minute with a working load of 10,000 pounds. Each crane is equipped with a fifty horsepower hoisting motor and a fifty horsepower training motor, both motors being compound wound, type CBC, Form A, R.P.M. 415, manufactured by the General Electric Co., as also are the controllers and the magnetic brakes. The operating machinery for the cranes is located on the third deck to insure freedom from mechanical injury and moisture grounds.

TOWING ARRANGEMENTS.

A two and a half inch diameter steel wire rope hawser (passed around the barbette of No. 4 turret) and connecting shackles, etc., are provided for towing.

ANCHOR HANDLING.

The windlass was constructed by the Hyde Windlass Co., of Bath, Maine. It is of the vertical type operated by two electric motors and consists of three wild cats on vertical shafts with locking gears, friction band brakes and bed plates, each vertical shaft having a worm wheel at its lower end, driven by a single worm on the extension shaft from the motors.

The motors themselves are General Electric, Type MPC, Form LP, H.P. 175, compound wound. The controls of the motors are of the distant master contactor type, manufactured also by the General Electric Co.

DECK WINCHES.

Four electric driven, compound geared, deck winches are installed, two forward and two aft on the weather deck. These winches were manufactured by the Hyde Windlass Co., of Bath, Maine, and are driven by General Electric Company's type BCB 25, Form A, compound wound 400 R.P.M. motors. The magnetic brakes, drum controllers and contactor controllers were also manufactured by the General Electric Company.

CAPSTANS.

An electric driven, compound geared capstan, built by the Hyde Windlass Co., is located on the main deck aft of No. 4 turret. The machinery is located on the third deck and is driven by a General Electric 45 horsepower motor with drum and contactor controllers also built by the same company.

KITE BALLOON WINCH.

A kite balloon winch is installed aft on the third deck—the cable lead coming down from a connecting tube leading to the quarter deck. The winch is built by the N. C. L. Engineering Corporation, of Providence, R. I., and is of the indirect spooling type designed to handle observation balloons through a wire at a wide range of speeds—725 to 1,900 R.P.M.—and hence under widely fluctuating torques. The winch is driven by a General Electric Co. 50 horsepower motor.

STEERING GEAR.

The steering gear consists of an electric-hydraulic system having two main hydraulic plungers, each plunger working on a pair of opposed hydraulic cylinders whose connecting rods

connect to the rudder cross head. The cross head is keyed to the upper end of the rudder stock. The rudder has an area of approximately 445 square feet.

Oil is pumped from one side to the other of the main hydraulic cylinders by two variable stroke pumps each consisting of a size 50, type K—Waterbury Tool Co. hydraulic speed gear. The two pumps working together are direct connected through flexible couplings to the ends of the shaft of a 125 horsepower, Type MPC, Form LD, 400 R.P.M. shunt wound General Electric Co. motor. A duplicate set of main motors and pumps is provided as a reserve. Either set can be connected to the system by the operation of valves at an oil manifold from which four pipes lead aft, one each of which connects to each ram cylinder.

The amount of oil pumped and the direction of its flow is dependent on the movement in degree and direction of the Waterbury Pump Tilting Box, which, in turn, is moved by a control shaft.

This control shaft is also connected to the follow-up mechanism driven from the rudder head which brings the pump tilting box back to neutral, hence stopping the rudder movement when the controller of the pilot motor actuating the crank mechanism is brought to the off position. A torque equalizing, or pressure control system, is incorporated in the follow-up mechanism to prevent (*a*) overloading or stalling of the motors, (*b*) to prevent the rudder taking charge or reversal of power. This pressure control is accomplished on the *Maryland* by providing a floating fulcrum on the lever actuated by the pilot motor which, through bell cranks, operates the tilting box of the oil pumps. This floating fulcrum is balanced between a spring and piston working within the torque equalizing cylinder, to which the system pressure is admitted. The design aims to be such that throughout the normal range of pressure no movement of this fulcrum will take place, but, with excessive pressures the pressure in the

torque equalizing cylinder will overcome the spring, allow the fulcrum to move and thereby reduce the angularity of the main pump valve plates and hence the amount of power transmitted by the oil motors.

The control shaft is operated by either one of a duplicate set of three horsepower, series wound, type CO, General Electric motors equipped with a type CR magnetic brake and control panel. This motor is started, stopped or reversed by a standard type, DL controller of the non-follow-up type (manufactured by the General Electric Co.), located at the various steering stations. In addition a "trick" wheel is fitted in the steering room for the operation of the control shaft by hand.

In the event of failure of the ship's main electric power, provision is made for both hand and auxiliary power steering. Briefly this system consists of an hydraulic pump, type 50, Waterbury Hydraulic pump of the same size and construction as those used in the main power drive fitted to be driven by any one of the following methods: (1) hand wheel; (2) a seven and a half horsepower auxiliary shunt wound motor, G. E. Type RC-11, Form AB, 80 volts, supplied with power by an 80-volt Exide Ironclad storage battery; (3) a five horsepower compressed air motor of the Chicago Pneumatic Tool Co., size 6AK. Local hand control only is provided for emergency use.

LAUNDRY.

The laundry is located on the main deck and is equipped with the following apparatus:

2 Pressing Machines, American Laundry Machinery Co., Type M.
1 Body Ironer, American Laundry Machinery Co., Type M.
1 Flat Work Ironer, American Laundry Machinery Co., Type M.

1 Collar Presser, The Reed Mfg. Co., Type W.
2 Extractors, American Laundry Machinery Co., Type M.
1 Sterilizer, American Sterilizer Co.
2 Washing Machines, American Laundry Machinery Co., 42 inches × 72 inches.
1 Steam Starching Cauldron, Navy Type, 17 inches × 14 inches.
1 Steam Drying Room, American Laundry Machinery Co.
1 Tumbler Drier.
1 Ironing Table.
1 Soap Tank.
4 Tubs.

In the Crew's Galley are installed ten Navy Standard oil burning ranges of the A type, No. 44, manufactured by the Bramhalt Dean Co., while the officer's galley is equipped with five Navy Standard type A, No. 58 oil burning ranges built by the Sexton Stove Co., of New York. Two systems of atomization are fitted and are optional for use—air and steam. The oil tanks feeding the galleys are connected to the booster pump discharge, the oil being forced to the burners from these tanks, by air pressure. An electric driven, single stage air compressor, built complete by the Nashville Brake and Electric Co., of Milwaukee, is installed for supplying air to the ranges and incinerator at thirty-five pounds pressure.

The following additional equipment is installed in the crew's galley:

2 Coffee Urns—Navy Standard, 120 gallons.
4 Steam Kettles, Navy Standard, Steam jacketed, 80 gallons capacity.
1 Potato Masher—York Machine Co., Electric driven.
2 Potato Peelers—Electric driven, 30-pound type.
1 Meat Chopper—Royal type—Electric driven.
1 Ice Cream Freezer—40-quart capacity on main deck.
1 Ice Cream Freezer Box—80-gallon capacity on main deck.

4 Warming ovens located in serving room.
1 Dish Washing Machine—Electric driven located in general mess pantry.

BAKERY.

The following equipment is installed in the bakery:
2 Cake Mixers—Electric driven—Reed Mfg. Co.
1 Dough Mixer—Electric driven—Days Co.
2 Bake Ovens—Electric driven—Simplex Type No. 1.
2 Proofers—Navy Standard—Manufactured by Blickman Co.
2 Dough Troughs—portable.

INCINERATOR.

A garbage and refuse incinerator of the Jockers type, manufactured by the Newport News Shipbuilding and Dry Dock Company, is installed on the port side of the superstructure deck forward. The oil supply is taken from the crew's galley tanks on the upper deck. Air for atomization is supplied by the galley compressor on the main deck. A one-quarter H.P. electric driven blower is used for forced draft.

DRAINAGE SYSTEM.

The drainage system consists of a main drain, secondary drain forward and secondary drain aft. The main drain has no connection or by-pass to the forward secondary drain but a six-inch line connects the main and after secondary drain. The main drain consists of four independent units, two in each machinery space. One of these units is connected to the main circulating pump and one eight-inch main to the fire and bilge pumps; the connection to the main circulating pump consists of a single suction, the valve being interlocked with the sea suction to prevent opening both at the same time.

The forward secondary drain is connected to a direct connected electric driven centrifugal pump located in compartment A-109. This pump has a capacity of 750 gallons per minute against a seventy-foot head and is driven by a thirteen and a half horsepower, type CB 45, Form A, compound wound General Electric motor. The after secondary drain is connected with the shaft alley bilge pump which is an electric driven plunger type pump located in the port center thrust room and has a capacity of two hundred and fifty gallons per minute against a head of thirty feet. This pump is driven by a seven horsepower compound wound, Type RC 11, Form G1, General Electric motor.

For draining compartments not connected to the drainage system two submersible salvage pumps of alternating current type are furnished. The plans were furnished by Submersible and J. L. Motors, of Middlesex, England, built by the Triumph Electric Company. One unit consists of the following: A fifteen horsepower, 220-volt, 3-phase, 60-cycle, 1,800 R.P.M. A. C. motor direct connected to a four-inch centrifugal pump rated at 430 G.P.M. against a 75-foot head; motor compensator; special three-conductor cables, each 75 feet long; two reels for cable; one foot valve and strainer, 4 inches; 3 suction hose, 4 inches, each 8 feet long; one suction elbow; two canvas discharge hose, 4 inches, each 50 feet long; one wooden truck with rollers. Terminals for connecting the pump are distributed throughout the ship as necessary, but all are located behind armor. Power supply is furnished by two fifty K.V.A. motor generating sets.

FIRE SYSTEM.

The firemain is so designed that maximum protection behind armor both above and at the sides is secured. The loops in the machinery spaces, boiler rooms and storerooms are all below the third deck and are thoroughly protected. There are

two risers in the barbettes, one in barbette No. Two and one in barbette No. Three. The barbette risers have branches at each deck fitted with cut-out valves operated from outside the barbette just above the cross connection.

FIRE EXTINGUISHERS.

For extinguishing oil fires one or two extinguishers of the foam type together with necessary hose lines and steam connections are installed in each fireroom and machinery space. The mixture for these foam type extinguishers consists of two chemical solutions which, when combined, generate carbon-dioxide gas forming a heavy foam, which by blanketing the burning surface, extinguishes the fire. Portable pyrene (carbon-tetrachloridic) extinguishers are located in various parts of the ship. Provision is also made for flooding all magazines by the use of sprinkling pipes or direct through flooding valves.

SANITARY SYSTEM.

Outside of an independent flushing system for the crew's, firemen's, C. P. O.'s and mess attendants' toilets, the sanitary system is supplied from the firemain. For the independent system, two electric driven centrifugal pumps located in compartment A-109 are connected to a 9-inch sea valve. These pumps have a 5-inch suction from the sea and a 5-inch discharge with a capacity of 500 G.P.M. against a head of 50 feet. Emergency connections to these pumps are fitted to the forward secondary drainage pump and fire main.

FRESH WATER SYSTEM.

Three main fresh water tanks are fitted in compartments A-114, A-115 and A-116—that in A-114 having a capacity of 88 4/10 tons and the others 43 4/10 tons apiece, a total of 175.2 tons. These tanks are filled by a 2½-inch distiller main

from the evaporator room with connections on the ship's side for obtaining water from outside sources. A 400-gallon tank is installed for providing the crew's and firemen's washrooms which is direct connected to the distilling plant as well as the fresh water delivery main from the ship's tanks. The fresh water delivery main is fed by either one of the two fresh water pumps located in compartment A-109. These pumps have a capacity of 200 G.P.M. against a 70-foot head and are driven by a compound type CB 45, form A, 12 H.P., 1,800 R.P.M. General Electric motor. These fresh water pumps take suction from the main fresh water tanks and deliver water by suitable piping to numerous outlets throughout the ship. These fresh water lines are fitted with recording meters such that an analysis may be made of the quantity of water used and the purpose therefor.

COMPRESSED AIR SYSTEM.

Two Ingersoll-Rand torpedo air compressors are installed in the torpedo room forward. These compressors have a capacity of 50 cubic feet per hour at 3,000 pounds per square inch, having a speed of 350 to 400 R.P.M. They are direct connected to a type MPC, Form B 98, compound wound, General Electric motor. The air compressor itself is cooled with circulating water from a 2-inch sea chest or from the fire main.

Compressed air for the gas ejecting system is supplied from the following sources:

(*a*) Any one of four air compressors, all alike, installed on the third deck, outside of the machinery spaces.

(*b*) Either one of the two machinery space, steam driven air compressors.

(*c*) From the torpedo room accumulator.

The third deck air compressors are all identical, Worthington-Blake-Knowles, size 15 inches × 12¾ inches × 7 inches, driven by MPC, Form LD, 45 H.P., compound wound,

300 R.P.M., General Electric motors. The compressors are geared, two-stage, vertical type with a capacity of 175 cubic feet free air per minute at 150 pounds per square inch pressure. They are fitted with accumulator tanks arranged for automatic starting and stopping. The Westinghouse steam driven compressors, one in each machinery space, are 11 inches × 11 inches × 12 inches and have a capacity of 45 cubic feet of free air per minute at 150 pounds pressure and are connected to a reservoir tank. Any one of the four third deck Blake-Knowles compressors or the two Westinghouse steam driven compressors in the engine rooms can be used through a pneumatic main to supply air to the foundry, forge, pneumatic tools, pneumatic tube system, gas ejecting system or for the air motor of the steering engine.

VENTILATION AND HEATING SYSTEM.

To provide ventilation throughout the ship 72 supply and exhaust ventilating sets are installed, each set consisting of a centrifugal, multivane type fan direct connected to an electric motor, piped to a system of ducts from which branches are led to the numerous compartments requiring artificial ventilation. In the various living spaces and offices throughout the ship where heating is required, the air from the fans is passed through steam heated coils before delivery to the air ducts. These heater coils are so arranged that either one, two or three coils may be used. The temperature can furthermore be regulated by the operation of a by-pass control damper. Means are also provided by which the humidity may be regulated through the operation of a hygrostat admitting live steam into the ventilating air. In some spaces, such as the Admiral's, Captain's and Wardroom country the electric heaters of the Simplex type are installed. Furthermore, the emergency staterooms fitted in the bridge structure employ steam radiators for heating.

MAIN PROPELLING MACHINERY.

The main propelling machinery of the *Maryland* consists of two direct connected turbine driven alternators, two main condensers, four two-phase induction motors, two booster generating sets for excitation, supplied with power by any one of six 300 K.W. geared turbine generating sets, motor and steam driven auxiliaries, main switching equipment, exciter and auxiliary switchboard, ventilating blowers for the main motors, control panels and wiring for electric auxiliaries together with necessary wiring for connecting main generators and motors with the switching equipment. The weight of the main propelling machinery as listed above is about 1,800 tons. The general arrangement of machinery is shown in Figures 1 and 2.

MAIN TURBINES.

The main turbine arrangement is shown in Figures 3 and 4. The turbine is a General Electric Curtis type, Form W, 10-stage, type HT, 11,000 K.W., 2,065 R.P.M., 250 pounds steam pressure. It has ten wheels, all having a single row of blading except the first, which has two. The steam enters the turbine through the strainer to a 14-inch throttle valve, under the control of an emergency governor, thence to the controlling valves, which in turn are under the control of the operating governor. The quantity of steam admitted is regulated by the operating governor which operates the pilot valve of the hydraulic cylinder by an arrangement of floating levers which are in turn connected to the governor lever. The pilot valve admits oil under pressure to the hydraulic cylinder and thereby moves a piston and rack up or down and thus closes or opens the controlling valves by means of a horizontal cam shaft. The steam chest has ten controlling valves with ports leading from each valve to a group of first stage nozzles. These controlling valves are operated by cams so arranged on the cam shaft that the valves are raised or lowered in suc-

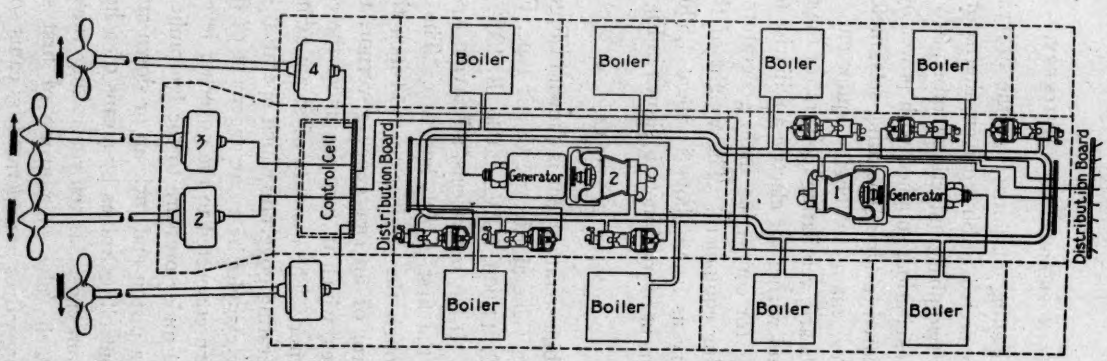
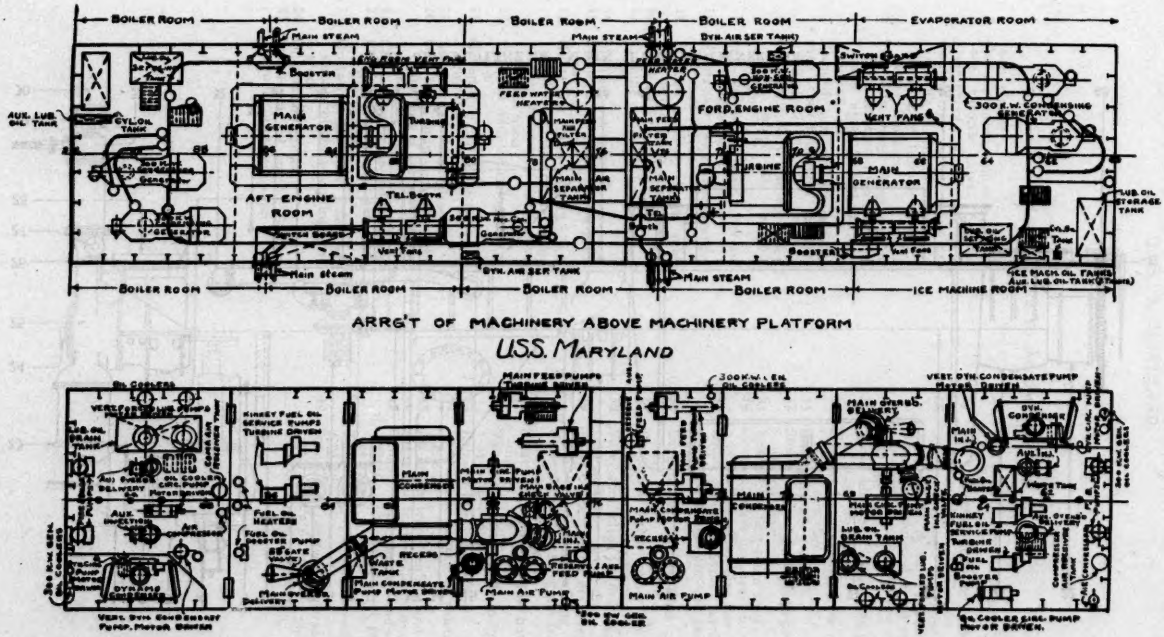


FIG. 1.—DIAGRAMMATIC ARRANGEMENT OF PROPULSION MACHINERY.



ARRG'T OF MACHINERY ABOVE MACHINERY PLATFORM

U.S.S. MARYLAND

FIG. 2.

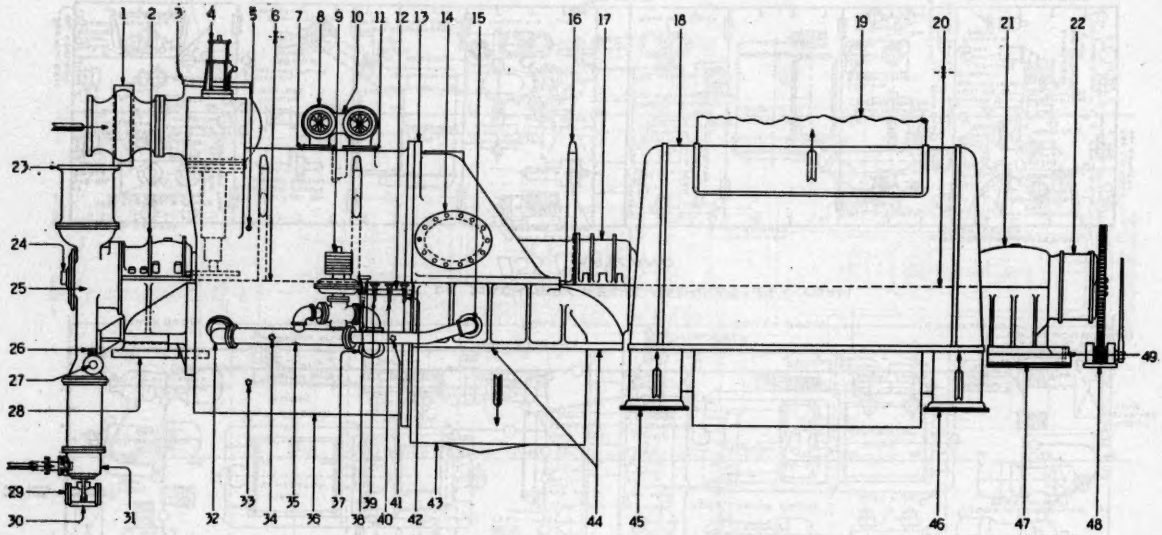


FIG. 3.—OUTLINE OF MAIN TURBINE-GENERATOR.

Nomenclature to Accompany Fig. 3.

1 14-in. strainer
2 Cap for turbine-end standard
3 14-in. throttle valve. See assembly of 14-in. throttle valve
4 Controlling valves. See assembly of controlling valve
5 Pipe plug in turbine shell for taking clearance
6 9-ft. 8-in. head room needed from horizontal joint to remove upper half of the turbine shell
7 Flanged auxiliary steam inlet to the fifth-stage bucket-wheel
8 8-in. angle body hold open stop check valve. See assembly of valve
9 4-in. unloading valve. See assembly of valve
10 Emergency valve. See arrangement of valve
11 Flanged auxiliary steam inlet to eighth-stage bucket wheel
12 Valve board
13 Turbine exhaust hood—upper half
14 Manhole cover for 13
15 Steam connection to low-pressure packing
16 Guide pin for 6 and 13. Assemble in place before lifting upper halves of shell and exhaust hood
17 Caps for middle-bearing standard
18 Generator. See assembly of generator
19 Lagging and vent shutters
20 6-ft. 10-in. head room from the center line of the unit needed to remove the stator
21 Cap for generator-end standard
22 Cover and guard for generator collector rings
23 Operating governor. See assembly of the operating governor
24 Oil pressure gauges
25 Spiral gear oil pump. See assembly of the oil pump
26 Oil baffler. See assembly of baffler
27 Oil connection to the coolers and bearings
28 Sliding support. Maximum movement, $\frac{1}{4}$ in. Flexibility must be provided in all pipe connections made to this end of the unit
29 Shock absorber. See assembly of shock absorber
30 Support to be provided for the shock absorber
31 Casing for 25
32 Steam connection to high-pressure packing
33 Drain from first-stage shell
34 Connection for live steam to the pressure packing pipe line
35 Drain connection from turbine head, and high-pressure packing
36 Lagging for turbine head and shell
37 Unloading valve connection to ninth-stage diaphragm
38 Valve connection for live steam to packing
39 Valve connection for drain from turbine head and packing
40 Valve connection for drain from unloading valve
41 Steam to the low-pressure packing
42 Valve connection for drain from first-stage shell
43 Exhaust hood—lower half
44 Foundation surface of exhaust hood
45 Air intake trap—turbine end
46 Air intake trap—collector end
47 Generator-end standard and sole plate. See assembly of generators
48 Turning device. See assembly. of device
49 Insulation between generator-end standard and sole plate. See details of insulation

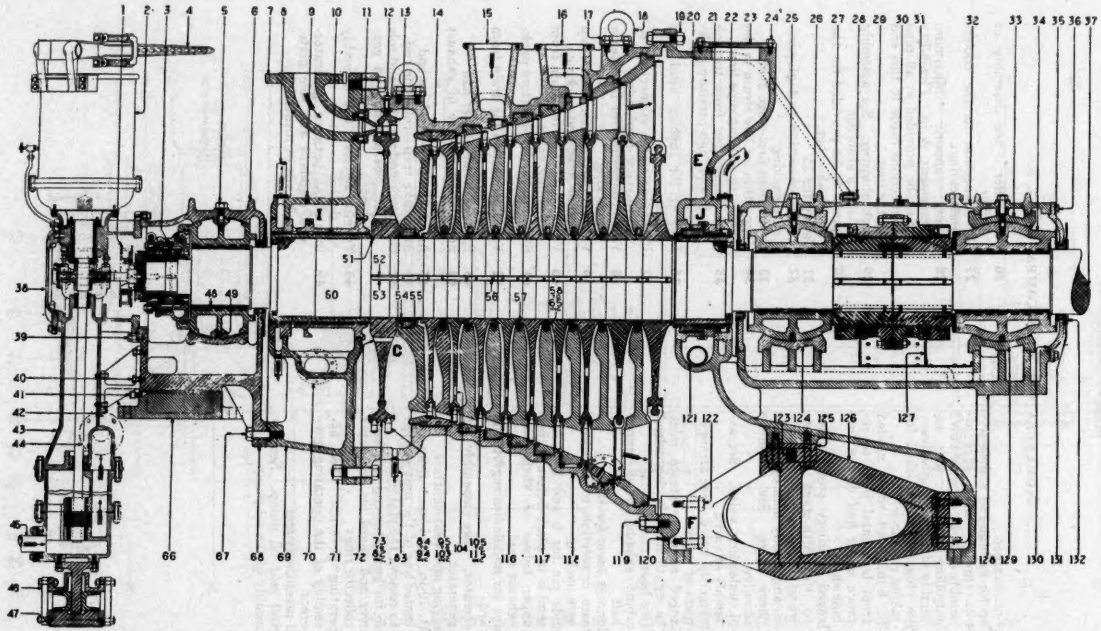


FIG. 4.—ASSEMBLY OF 10-STAGE FORM "W" HORIZONTAL CURTIS STEAM TURBINE FOR DIRECT CONNECTION TO ALTERNATING-CURRENT GENERATOR.

Nomenclature to Accompany Fig. 4.

1 Emergency governor. See assembly of emergency governor
2 Operating governor. See assembly of operating governor
3 Thrust bearing. See assembly of thrust bearing
4 Operating governor lever
5 Dowel bolt for 6 and 48
6 Cap for 68
7 Turbine head—upper half
8 Vent pipe to atmosphere, high pressure packing
9 Pipe plug for measuring steam pressure and temperature of high-pressure packing
10 High-pressure packing. See assembly of packing
11 Balancing weights
12 Tap bolt for 13 and 14
13 Intermediate segment with stationary buckets
14 Turbine shell—upper half
15 Flange for connecting the auxiliary steam pipe line to the fifth-stage diaphragm
16 Flange for connecting the auxiliary steam pipe line to the eighth-stage diaphragm
17 Stud bolt for 14 and 18
18 Lifting eye for 14
19 Bolt and nut for 14 and 21
20 Low-pressure packing. See assembly of the packing
21 Exhaust hood—upper half
22 Vent pipe to atmosphere, low-pressure packing
23 Cover
24 Stud bolt for 21 and 23
25 Bearing ring for 26 and 27
26 Cap for 25 and 128
27 Turbine-middle bearing
28 Bolt and nut for 26 and 29
29 Cap for 128
30 Tapped hole for lifting eye
31 Flexible jaw coupling
32 Cap for 33 and 128
33 Bearing ring for 32 and 34
34 Generator-middle bearing
35 Oil guard
36 Tap bolt for 32 and 130
37 Generator shaft
38 Gauge board
39 Stud bolt for 42 and 68
40 Stud bolt for 42 and 68
41 Grease cup for 66
42 Bracket for 43 and 68
43 Gear casing for oil pump and operating governor drive
44 Oil outlet flange
45 Oil inlet from supply tank
46 Shock absorber. See assembly of the absorber
47 Foundation for 46 to be supplied by the shipbuilder
48 Turbine-end bearing
49 Tapped hole for lifting-eye
50 Turbine shaft
51 Locking key
52 Key for 50 and 73
53 Auxiliary key for 50, 52 and 73
54 Packing sleeve for the shaft
55 Second-stage diaphragm packing ring
56 Keys for 50 and 74 to 82 inclusive
57 Axial crushing pins for 74 to 81 inclusive
58 to 65 inclusive Diaphragm packing rings
66 Sole plate for 68
67 Tap bolt for 68 and 69
68 Turbine-end standard
69 Turbine head—lower half
70 Flanged opening for steam to 10
71 Bolt and nut for 69 and 118
72 Hole for inserting bolts for removing 73
73 to 82 inclusive Bucket wheels
83 to 94 inclusive Buckets
95 to 103 inclusive Nozzle diaphragms, upper and lower half. See assembly of nozzle diaphragm
104 Equalizing holes
105 to 115 inclusive Bucket covers
116 Axial crushing pins for 97 to 101 inclusive
117 Radial crushing pins for 95 to 103 inclusive
118 Turbine shell—lower half
119 Tap bolt for 118 and 128
120 Pipe plug. Measure vacuum and temperature here
121 Steam passage for sealing the low-pressure packing
122 Tap bolt for 126 and 128
123 Tapped hole for jacking bolt
124 Bearing ring—lower half for 27 and 128
125 Filler block for 126 and 128
126 Strut for 128
127 Coupling guard
128 Exhaust hood—lower half
129 Bearing ring—lower half for 34 and 128
130 Oil deflector holder
131 Air shield
132 Oil deflector

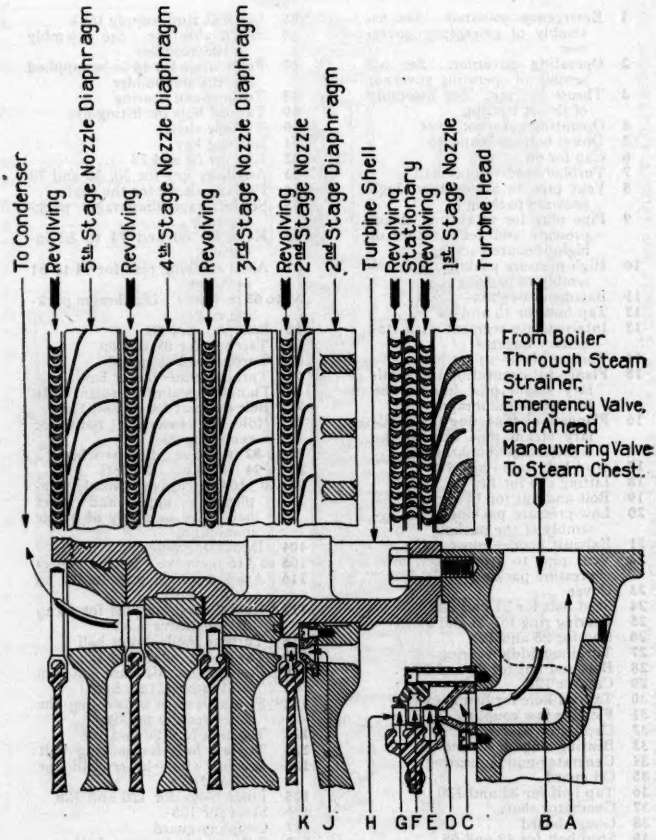


FIG. 5.—DIAGRAM SHOWING PATH OF STEAM THROUGH CURTIS STEAM TURBINE.

cession, admitting such quantity of steam as to maintain a predetermined speed. Referring to Figure 5 the steam enters the chest A and after passing through bowl B expands through the first stage nozzle C entering the first row E of revolving buckets of the first stage wheel at D; thence, it passes through the stationary buckets F which reverse its direction and redirect it against the second revolving row G of the same wheel. This constitutes the performance of the steam in one stage or pressure chamber. Having entered the first row of buckets at D with relatively high velocity, it leaves the last row G of the first stage with a relatively low velocity, its energy between the limits of the inlet and storage pressure from C to D having been extracted in passing through the first stage to the nozzles of the first diaphragm. It has, however, a large amount of unexpended energy, since the expansion from C to D covers only a part of the available pressure range; the expansion process is therefore repeated in a second stage. The steam, having left the buckets of the first stage wheel and having had its velocity greatly reduced, reaches a second series of nozzles J for the second stage. Through these nozzles the steam again expands from the first stage pressure to a lower pressure, again acquiring relatively high velocity in its expansion through these nozzles, leaving them and impinging upon and passing through the moving buckets of K of the second stage wheel. This and the following stages have but one row of moving buckets; hence, no stationary buckets are necessary to reverse the direction of the steam and it passes directly to the nozzles of the following stage where precisely the same process is repeated. After the steam is passed through the last stage practically all its available energy has been used and it flows directly to the condenser.

Provision is also made for admitting through spring loaded valves auxiliary exhaust steam and the overflow steam of the pressure packings to either the 5th or 8th stage nozzle diaphragms. The auxiliary steam enters these stages of the

main turbine at pressures and temperatures that are about equal to the pressures and temperatures of the stages into which it enters.

The first stage nozzle segment is supported in a recess of the upper half of the turbine head and is secured by tap bolts. The nozzle partitions are equally spaced and cast integral with the segment which has ten groups of openings registering with corresponding openings or bowls in the turbine head. All the nozzle groups are under control of the operating governor which regulates the controlling valves by means of hydraulic operating mechanism.

The bucket wheels are heat treated steel forgings machined over their entire surface, pressed and keyed to the shaft. To obtain a good fit between wheels and the shaft the bores of the hubs have varying diameters increasing from the first wheel and reaching the maximum at the tenth. The rims of the wheels are machined to receive a single row of buckets excepting the first which carries a double row of blading. The buckets of the first eight wheels are formed with a dove-tailed shaped tenon while those of the remaining two wheels are of the inverted type, the buckets being carefully finished and polished before assembling. The ninth and tenth wheels are machined to take the inverted type of bucket, the other wheels being grooved to take the dove-tailed type. A notch is cut in the wheel rim for assembling the blading and this is afterwards closed by a notched block secured by rivets. Equalizing holes with well-rounded edges are equally spaced in the disc to relieve the pressure on the steam admission side under abnormal conditions, such as buckets being clogged with scale or very wet steam under priming conditions.

All wheels except the first and tenth are drilled at the hub to take four brass crushing pins on the steam exit side to compensate for expansion when the shaft is thoroughly heated. The first stage wheel has twelve holes drilled and fitted with steel bushings near the outer rim for use in attaching balancing

weights. These holes are fitted with bushings hot riveted in position, the shoulders swaged and peened tightly against the wheel web; these bushings are then drilled and tapped for the balancing weights. Wherever necessary these weights are screwed into bushings but do not extend beyond the disc surface more than twice the diameter of the weight. The balancing weight is secured by punching the metal of the bushing in a slot cut in the weight. The balancing weights are accessible through a man hole in the upper half of the turbine head. The intermediate segment for the reception of the stationary buckets of the first wheel consists of three steel sections with a dove-tail groove for these stationary buckets. It is securely bolted to the upper half of the turbine shell by sherardized tap bolts extending through the shell. The nozzle partitions are of nickle steel, cast integral with the diaphragm. The diaphragms are in halves, split horizontally, and fitted at the joints with non-corrodible radical keys to maintain alignment of the halves and prevent steam leakage. The groove is machined in the circumference of the halves to take the diaphragm packing sleeves.

As the steam expands from stage to stage its volume rapidly increases and hence greater area for steam passage must be provided which is accomplished by increasing the height of the buckets and the number and area of nozzles for the various stages. The nozzles of all diaphragms of the turbine are equally spaced on the complete circumferences of the diaphragm. The upper halves of the 5th and 8th stage nozzle diaphragms are drilled through the outer rim to admit the auxiliary exhaust steam to the bucket wheels of these stages. The outer rim of the lower half of the 9th stage nozzle diaphragm is drilled to allow the overflow steam of the pressure packings through the unloading valve to enter 9th stage bucket wheel.

The diameter of the various diaphragms is such that a radial clearance of approximately 1/16 inch exists between the

periphery of the diaphragm and that of the shell. The axial clearance between the diaphragm and the shell of 40/1000 of an inch for the diaphragms of the 4th to 8th stages, inclusive. The 9th and 10th stage diaphragms have no axial clearance as a steam tight joint is made by narrow chamfered flanges on the outer periphery of the diaphragm which fits into a corresponding groove in the shell. The 2nd and 3rd stage diaphragms set in the same groove and have an axial clearance of 10/1000 inch to allow for expansion. The first stage of the turbine drains through a small pipe in the lower half of the turbine shell. The drainage of the 2nd to the 9th stages flows along through a series of grooves in the bottom halves of the diaphragm till it reaches the exhaust casing.

The turbine shaft is of forged nickle steel machined over its entire surface and "stepped" toward the exhaust end for the reception of the wheels and packing sleeves. Collars are turned on the shaft where it forms a journal for each of the main bearings to prevent oil running along the shaft.

Axial keyways are provided for reception of keys for the thrust collar, the diaphragm packing sleeves, the wheels and the flexible jaw coupling. The keys are individually fitted and are provided with tap holes for the use of jacking bolts when removing them from the shaft. Radial tap holes are provided on the head end of the shaft for the tap bolts of the emergency governor. Tap holes are also provided in the shoulders near the bearing collars for securing the packing sleeves of the high and low pressure labyrinth packings. The head end of the shaft is also slotted, bored and threaded to take the nut of the thrust bearing and the worm shaft connection of the spiral gear driven oil pump and governor drive. The axial bore of the main shaft is crossed by a radially drilled hole for furnishing lubrication to the collar of the thrust bearing through the worm shaft connection.

The axial thrust of the turbine rotor is absorbed by an adjustable thrust bearing on the end of the turbine shaft,

between the emergency governor and the turbine end bearing. This thrust collar is comparatively small and the clearance between it and the thrust bearing rings is also small, inasmuch as this type of turbine exercises a comparatively small thrust except in cases of priming. The thrust collar, consisting of a steel plate shrunk and keyed to the shaft, runs between two thrust bearing rings split at the horizontal joint and fitted with spacing pins. These thrust collars are adjustable through a worm shaft and wheel. Squealer rings are fitted between the sleeve which fits between the thrust collar and the nut for securing the same, so that, should there be any tendency of the turbine wheels when running to move the shaft axially toward the generator when the thrust rings have worn abnormally, the squealer rings will rub and give notice of trouble. Adjustable shims allow for proper adjustment of distance between these rings.

TURBINE THROTTLE VALVE. FIGURE 6.

The turbine is fitted with a 14-inch balanced throttle and trip valve, the trip feature of which is released either by a hand lever or through a rocker connected to the emergency tripping device of the overspeed governor, the hand trip having an extension to the main operating control room. The valve will close through operation of the emergency governor when the turbine speed is increased to about 10 per cent above normal. An emergency valve to regulate the amount of steam passing from the auxiliary apparatus to the lower pressure stages of the main unit is placed in the auxiliary steam line on the port side of the upper half of the turbine shell just outboard of the two 8-inch angle body stop check valves. This valve consists of a "Y"-shaped casing containing a butterfly valve on one side and a clapper valve in the center. The butterfly valve feature is connected to the hydraulic governor mechanism so that when any movement of the hydraulic governor mechanism tends to slow the speed of the turbine

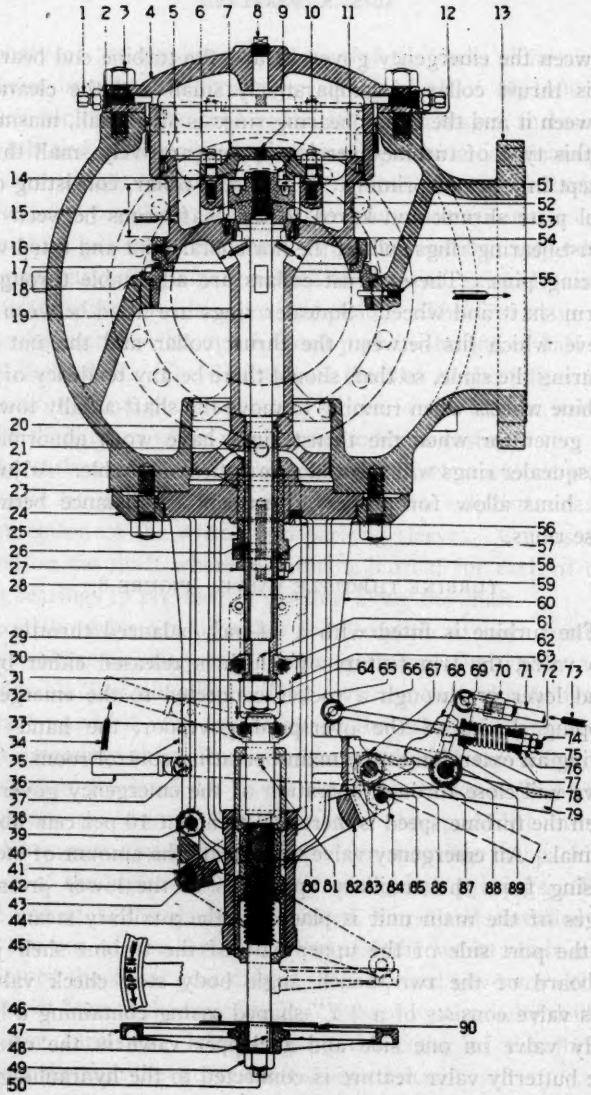


FIG. 6.—ASSEMBLY OF 14-INCH BALANCED THROTTLE AND TRIP VALVE (U. S. S. "MARYLAND").

Nomenclature to Accompany Fig. 6.

1 Throttle screw and check nut for 4
2 Packing—graphite lubricated. Avoid those containing rubber
3 Stud with nut for 4 and 13
4 Cover with cylinder
5 Piston ring
6 Collar stud with nut and cotter pin for 7 and 11
7 Spool cap
8 Pipe plug. Enter eyebolt here for lifting 7 and attach pressure gauge here for measuring pressure in balancing chamber
9 Tapped hole for eye-bolt
10 By-pass valve
11 Spool
12 Steam-tight joint, between 4 and 13
13 Valve body
14 Pin for 10 and 19. Rivet over
15 Spool lift = 4 in.
16 Socket screw for 11 and 17. Peen metal of 17 into slot of screw
17 Seat face ring
18 Seat ring
19 Spindle—upper
20 Below seat drain. Remove pipe plug
21 Packing ring
22 Packing—graphite lubricated. Avoid those containing rubber
23 Steam-tight joint, between 13 and 24
24 Solid clamp ring for 60
25 Large gland
26 Drain from 25
27 Small gland
28 Closing sleeve
29 Steam joint shoulder between 19 and 28
30 Key for 28 and 63
31 Steam-tight joint, between 28 and 64
32 Clearance of $\frac{1}{8}$ in. between 19 and 45
33 Oil hole in 63 for lubricating upper collar of 45
34 Direction to lift 36, to trip valve by hand
35 Shackle—to connect to the ship's control connections
36 Hand lever
37 Pin for 35 with cotter pin and washer
38 Oil hole in 60
39 Pin for 41 and 60 with cotter pin and washer
40 Name and reference plate. "Do not paint over"
41 Link
42 Oil hole in 82
43 Pin for 41 and 82 with cotter
44 Sliding nut
45 Screw spindle—lower
46 Caution plate. "Do not paint over"
47 Handwheel
48 Key for 45 and 47
49 Nut for 45 and 47
50 Cotter for 45 and 49
51 By-pass valve lift = $\frac{1}{4}$ in.
52 Steam joint shoulder between 10 and 19
53 Screw for 11 and 54
54 By-pass valve seat ring
55 Socket set screw for 13 and 18
56 Pipe plug for drain below seat
57 Packing—graphite lubricated. Avoid those containing rubber
58 Stud and nut for 25 and 27
59 Stud and nut for 24 and 25
60 Yoke
61 Key for 19 and 28
62 Cap screw for halves of 63
63 Split coupling
64 Nut for 19 and 28
65 Lock nut for 19 and 64
66 Pin with cotter for 60 and 74
67 Oil hole in 84
68 Bell crank trip lever
69 Pin with cotter for 70 and 78
70 Clevis connection for 71 and 78
71 Connecting rod of the emergency tripping device
72 Compression spring
73 Direction of movement of 71 to trip the valve
74 Spring rod
75 Lock nut for 74 and 76
76 Nut for 74 and 77
77 Spring guide
78 Lever
79 Slotted opening in 36 for connecting to ship's control connections
80 Cap screw for 44 and 82
81 Oil holes in 44
82 Trip lever
83 Key for 84 and 86. Riveted over at both ends
84 Trip hook
85 Pin for 82. Riveted over
86 Trip shaft with nut and washer
87 Oil hole in 60
88 Bell crank shaft, with nuts and washer
89 Key for 68, 78 and 88. Riveted over at both ends
90 Screw for 46 and 47

down, the butterfly valve closes against the incoming steam, cutting off the available opening and hence the amount of steam passing to the lower pressure stages and vice versa. The clapper valve feature of the emergency valve is connected by a series of rods and levers to the emergency tripping device which in turn may be tripped by the emergency governor, by hand trip lever, by throttle trip or by the failure of the hydraulic pressure. When any of these trip devices cause the emergency trip to function, auxiliary steam is shut off the 5th and 8th stages.

OPERATING GOVERNOR.

The function of the operating governor shown in Figure 7 is to regulate the turbine speed under load variations. The governor accomplishes this through the hydraulic mechanism actuating the controlling valves and thereby controlling the quantity of steam supplied to the turbine, which in turn determines the speed at which the turbine runs. The governor is of the centrifugal type mounted on the upper end of the vertical governor drive shaft and is adjustable to operate between the turbine speeds of 500 R.P.M. and 2,065 R.P.M. at actual governor speeds of 108 R.P.M. and 612 R.P.M. The action of the governor depends on a balance between forces exerted by the spring and the centrifugal effort of the weights. When the turbine is at rest or when running up to a speed where the governor operation commences, the weights are in a vertical position. As the speed increases the weights travel outward until the stops of the weights come in contact with those fixed at the bottom of the bracket. At this point the maximum controlled governor speed is reached. When travelling upward the weights press against the fulcrum blocks. The hook-shaped weight brackets impart a downward motion to the governor lever transmitted through the links, spring blade, and spindle and roller transmission bearings. The governor lever transmits its motion through a series of levers to the pilot valve of

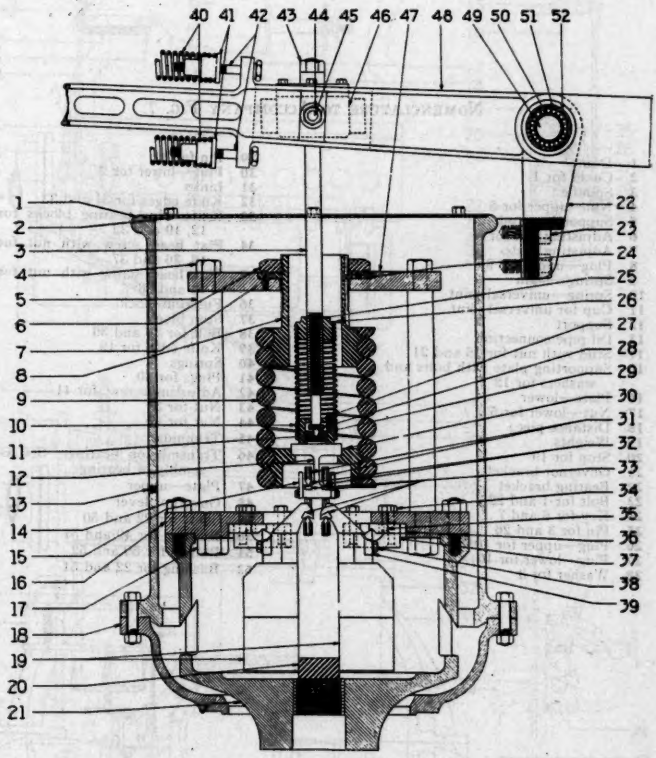


FIG. 7.—ASSEMBLY OF OPERATING GOVERNOR.

Nomenclature to Accompany Fig. 7.

1 Dome
2 Cover for 1
3 Spindle
4 Nut—upper for 5
5 Supporting stud
6 Adjusting nut for 8
7 Adjusting plate
8 Plug—upper for 9
9 Spring—main
10 Spring—universal joint
11 Cup for universal joint
12 Support
13 Oil pipe connection
14 Stud with nut for 16 and 21
15 Supporting plate with bolts and washers for 12
16 Plate—lower
17 Nut—lower for 5
18 Distance piece
19 Weights
20 Stop for 19
21 Governor bracket
22 Bearing bracket
23 Bolt for 1 and 22
24 Key for 6 and 7
25 Pin for 3 and 26
26 Plug—upper for 10
27 Plug—lower for 10
28 Washer for 3
29 Pin for 11
30 Plug—lower for 9
31 Links
32 Knife edges for 31 and 33
33 Knife edge bearing blocks for 12, 19 and 32
34 Flat head screw with nut for 16, 36 and 37
35 Flat head screw with nut for 16 and 36
36 Fulcrum block
37 Stop block
38 Bolt for 19 and 39
39 Knife edge for 19
40 Springs
41 Plugs for 40
42 Adjusting screws for 41
43 Nut for 3
44 Nut for 45
45 Trunnion
46 Transmission bearing. See assembly of bearing
47 Plate—upper
48 Governor lever
49 Shaft for 22 and 50
50 Bushing for 49 and 51
51 Rollers for 50 and 52
52 Bushing for 22 and 51

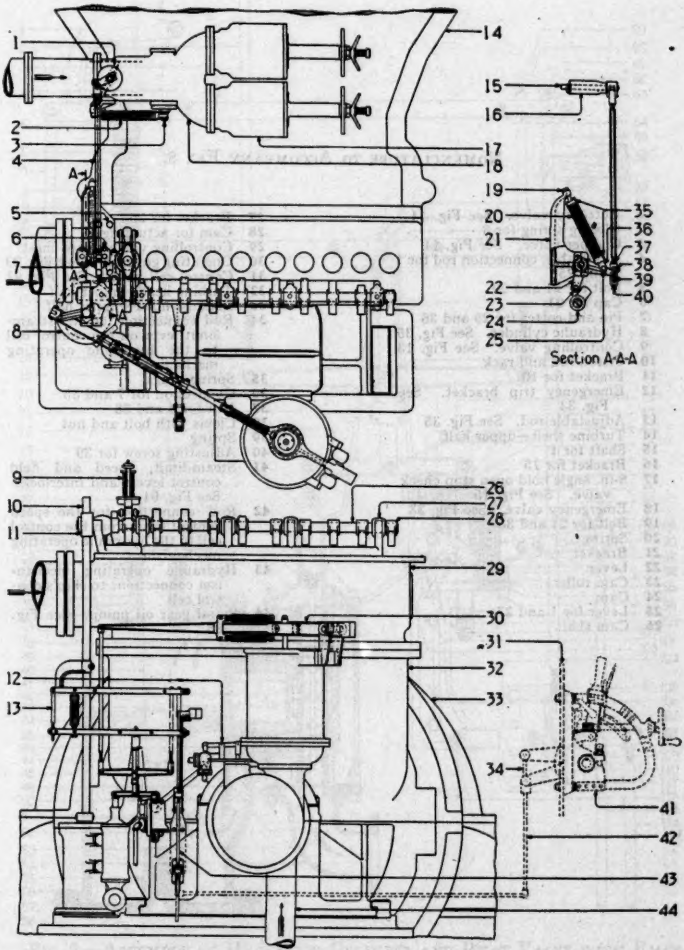
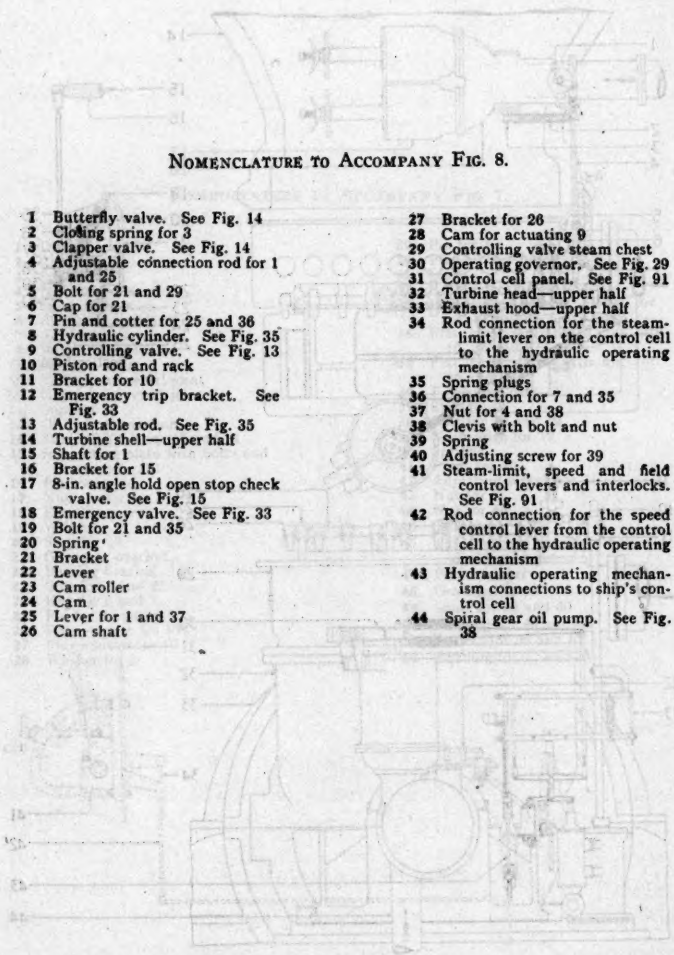


FIG. 8.—ARRANGEMENT OF HYDRAULIC OPERATING MECHANISM.



Nomenclature to Accompany Fig. 8.

1 Butterfly valve. See Fig. 14
2 Closing spring for 3
3 Clapper valve. See Fig. 14
4 Adjustable connection rod for 1 and 25
5 Bolt for 21 and 29
6 Cap for 21
7 Pin and cotter for 25 and 36
8 Hydraulic cylinder. See Fig. 35
9 Controlling valve. See Fig. 13
10 Piston rod and rack
11 Bracket for 10
12 Emergency trip bracket. See Fig. 33
13 Adjustable rod. See Fig. 35
14 Turbine shell—upper half
15 Shaft for 1
16 Bracket for 15
17 8-in. angle hold open stop check valve. See Fig. 15
18 Emergency valve. See Fig. 33
19 Bolt for 21 and 35
20 Spring
21 Bracket
22 Lever
23 Cam roller
24 Cam
25 Lever for 1 and 37
26 Cam shaft
27 Bracket for 26
28 Cam for actuating 9
29 Controlling valve steam chest
30 Operating governor. See Fig. 29
31 Control cell panel. See Fig. 91
32 Turbine head—upper half
33 Exhaust hood—upper half
34 Rod connection for the steam-limit lever on the control cell to the hydraulic operating mechanism
35 Spring plugs
36 Connection for 7 and 35
37 Nut for 4 and 38
38 Clevis with bolt and nut
39 Spring
40 Adjusting screw for 39
41 Steam-limit, speed and field control levers and interlocks. See Fig. 91
42 Rod connection for the speed control lever from the control cell to the hydraulic operating mechanism
43 Hydraulic operating mechanism connections to ship's control cell
44 Spiral gear oil pump. See Fig. 38

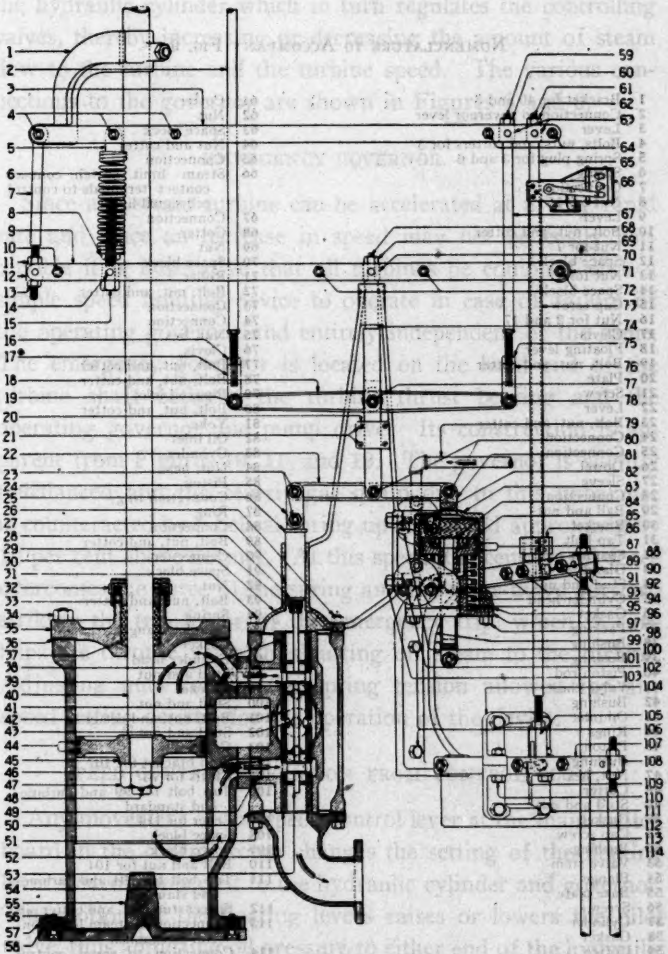


FIG. 9.—ASSEMBLY OF HYDRAULIC CYLINDER AND PILOT VALVE WITH EMERGENCY OIL CYLINDER.

Nomenclature to Accompany Fig. 9.

1 Bracket for 40 and 3
2 Connection to governor lever
3 Lever
4 Bolts, nuts, and cotters for 3
5 Spring plug for 3 and 6
6 Spring
7 Connection
8 Nut for 15
9 Lever
10 Bolt, nut, and cotter
11 Nut for 7
12 Space block
13 Nut for 7
14 Space block
15 Cap screw
16 Nut for 2 and 17
17 Clevis
18 Floating lever
19 Bolt, nut, and cotter
20 Plate
21 Screw
22 Lever
23 Bolt, nut, and cotter
24 Connection
25 Connection
26 Dowel
27 Sleeve
28 Connection
29 Ball and nut
30 Bracket
31 Tap bolt
32 Spanner nut
33 Gland
34 Stud and nut
35 Cylinder head
36 Gasket
37 Packing
38 Bushing
39 Bushing
40 Piston rod
41 Valve chest
42 Bushing
43 Oil inlet
44 Rings
45 Piston
46 Bushing
47 Nut, hex.
48 Cotter
49 Stud and nut
50 Gasket
51 Cap screw
52 Bushing
53 Valve stem
54 Flange
55 Base plate
56 Screw
57 Cylinder
58 Gasket
59 Cotter
60 Nut
61 Cotter
62 Nut
63 Space block
64 Nut and cotter
65 Connection
66 Steam limit switch, connect contact terminals to control cell signal lights
67 Connection
68 Cotter
69 Nut
70 Space block
71 Roller
72 Bolt, nut, and cotter
73 Connection
74 Connection
75 Nut, hex.
76 Clevis
77 Bolt, nut, and cotter
78 Bolt, nut, and cotter
79 Pipe, oil inlet
80 Bolt, nut, and cotter
81 Yoke
82 Oil inlet
83 Cylinder
84 Nut, hex.
85 Piston
86 Connection ring
87 Ring
88 Connection
89 Bolt, nut, and cotter
90 Connection
91 Space block
92 Nut
93 Bolt, nut, and cotter
94 Spring
95 Oil inlet, flange
96 Gasket
97 Cylinder head
98 Stud and nut
99 Bracket
100 Stud and nut
101 Bolt, nut, and cotter
102 Stud and nut
103 Cap screw
104 Stop bracket for 107
105 Clevis for 73
106 Tap bolt for 99 and turbine-end standard
107 Lever for 114
108 Space block
109 Space block
110 Bolt and nut for 104
111 Tap bolt for 104 and turbine-end standard
112 Spacer stud, nut, and cotter pin
113 Connection to steam limit control mechanism
114 Connection to speed control mechanism

the hydraulic cylinder which in turn regulates the controlling valves, thereby increasing or decreasing the amount of steam flow to the turbine and the turbine speed. The various connections to the governor are shown in Figures 8 and 9.

EMERGENCY GOVERNOR.

Since any steam turbine can be accelerated at a very rapid rate and since an increase in speed may not be easily perceptible it is imperative that all turbines be equipped with a simple speed limiting device to operate in case of failure of the operating governor and entirely independent of the same. The emergency governor is located on the head end of the turbine shaft between the turbine thrust bearing and the operating governor and pump drive. Its construction is apparent from Figures 10, 11, and 12. The governor is slightly unbalanced and the centrifugal strain due to this unbalance is counteracted by a helical spring up to a speed approximately 10 per cent above normal. At this speed, the centrifugal force overcomes the force of the spring and the rings move outward striking the trip finger of the emergency trip, which in turn trips the throttle valve thus cutting off steam to the turbine. Adjusting nuts control the spring tension allowed for any speed setting desired for the operation of the device.

SPEED CONTROL OPERATION FROM CONTROL ROOM.

Any movement of the speed control lever at the main switch board in the control room changes the setting of the floating lever system connected to the hydraulic cylinder and governor. The motion of the floating levers raises or lowers the pilot valve, thus admitting oil pressure to either end of the hydraulic cylinder and hence controlling the movement up or down of the hydraulic piston, thereby opening or closing both the butterfly valve and controlling valves through the rack and pinion connected to the hydraulic piston. The governor immediately

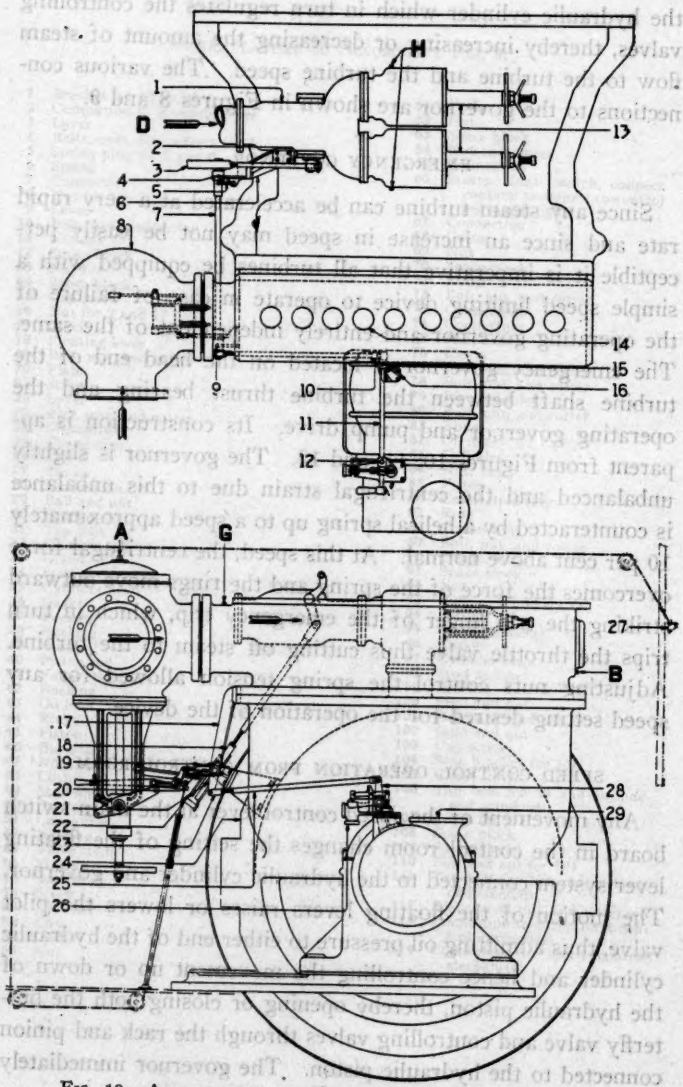
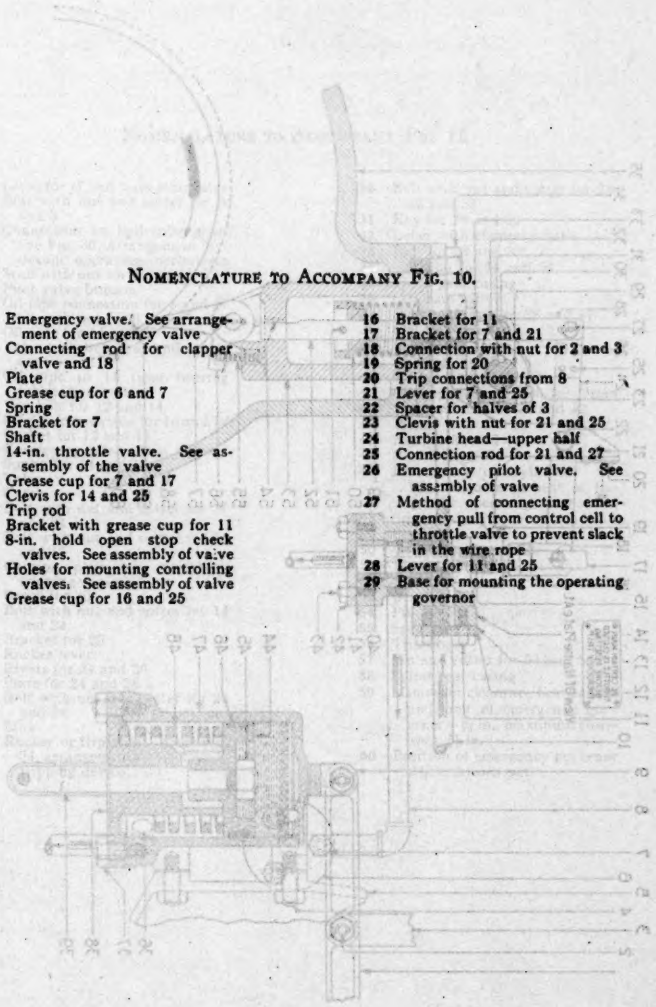


FIG. 10.—ARRANGEMENT OF EMERGENCY TRIPPING MECHANISM.



Nomenclature to Accompany Fig. 10.

1 Emergency valve. See arrangement of emergency valve
2 Connecting rod for clapper valve and 18
3 Plate
4 Grease cup for 6 and 7
5 Spring
6 Bracket for 7
7 Shaft
8 14-in. throttle valve. See assembly of the valve
9 Grease cup for 7 and 17
10 Clevis for 14 and 25
11 Trip rod
12 Bracket with grease cup for 11
13 8-in. hold open stop check valves. See assembly of valve
14 Holes for mounting controlling valves. See assembly of valve
15 Grease cup for 16 and 25
16 Bracket for 11
17 Bracket for 7 and 21
18 Connection with nut for 2 and 3
19 Spring for 20
20 Trip connections from 8
21 Lever for 7 and 25
22 Spacer for halves of 3
23 Clevis with nut for 21 and 25
24 Turbine head—upper half
25 Connection rod for 21 and 27
26 Emergency pilot valve. See assembly of valve
27 Method of connecting emergency pull from control cell to throttle valve to prevent slack in the wire rope
28 Lever for 11 and 25
29 Base for mounting the operating governor

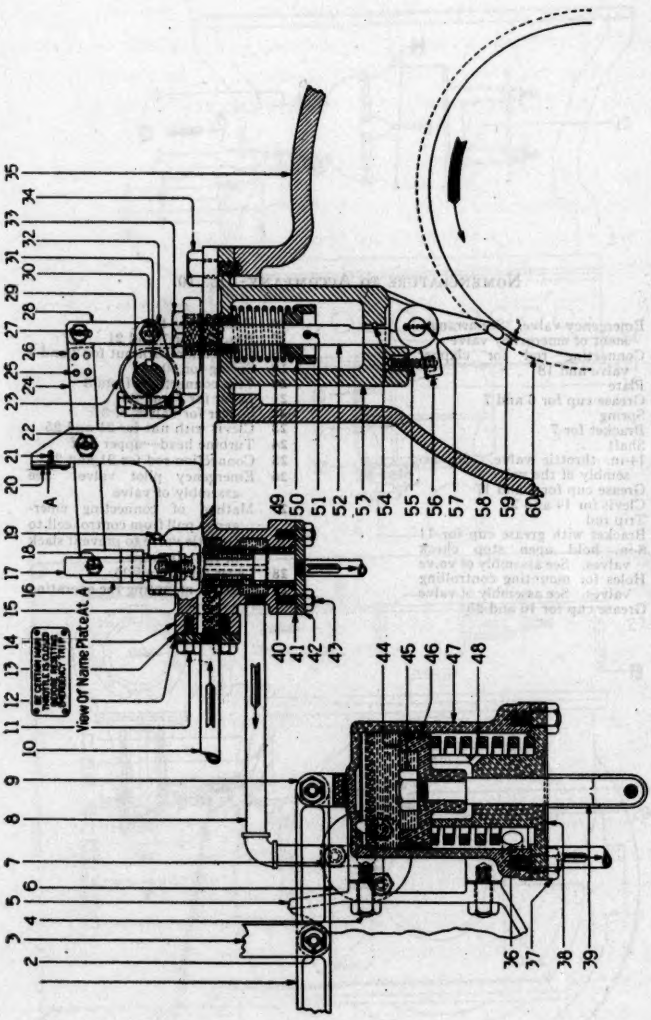
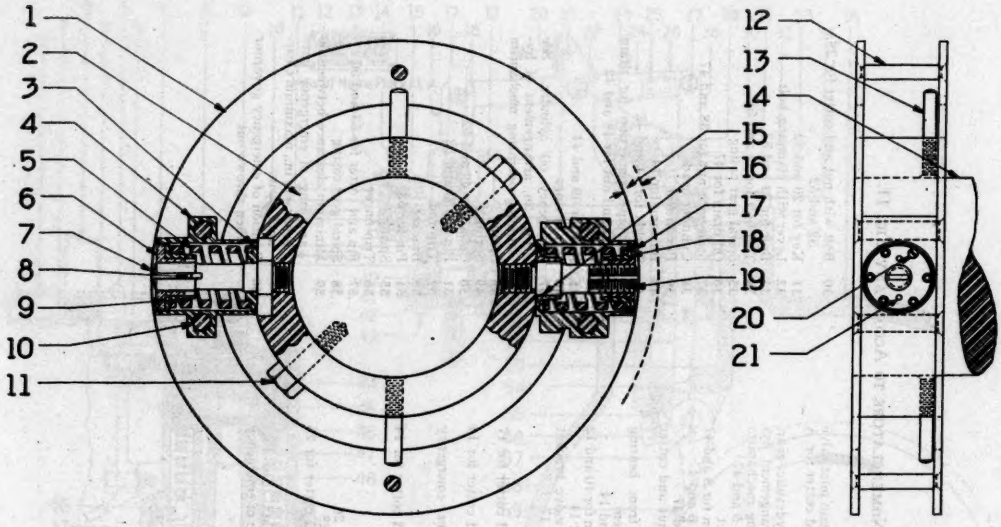


FIG. 11.—ASSEMBLY OF EMERGENCY TRIPPING DEVICE, PILOT VALVE, AND OIL CYLINDER.

Nomenclature to Accompany Fig. 11.

1 Lever for 47 and main pilot valve
2 Bolt with nut and cotter for A and 3
3 Connection to hydraulic gear. See Fig. 36, arrangement hydraulic operating mechanism
4 Stud with nut for 5 and 47
5 Pilot valve bracket
6 Oil pipe connection for 8 and 47
7 Stud with nut for 6 and 47
8 Oil pipe for 14 and 47
9 Yoke with bolt, nut and cotter for 1
10 Oil pipe to 14 from bearing lubrication system
11 Tap bolt for 12 and 14
12 Oil pipe connection for 10 and 12
13 Gasket for 12 and 14
14 Emergency pilot valve bracket
15 Spring for 14 and 17
16 Valve
17 Catch
18 Bolt with nut and cotter for 17 and 24
19 Bolt with nut and cotter for 14 and 17
20 Caution plate for emergency tripping device
21 Screw for 20
22 Bolt with nut and cotter for 14 and 24
23 Bracket for 29
24 Rocker lever
25 Rivets for 24 and 26
26 Plate for 24 and 28
27 Bolt with nut and cotter for 26 and 28
28 Link
29 Rocker or trip shaft. See Fig. 34, arrangement of emergency tripping device
30 Bolt with nut and cotter for 28, 32 and 52
31 Key for 29 and 32
32 Lever with clamping bolt
33 Bolt for 23 and 53
34 Tap bolt for 14 and 53
35 Spiral gear casing
36 Oil drain for 47
37 Stud with nut for 38 and 47
38 Cylinder head
39 Piston rod
40 Bushing for 14
41 Gasket for 14 and 42
42 Oil pipe connection for drain
43 Stud and nut for 14 and 42
44 Piston
45 Nut for 39 and 44
46 Piston rings
47 Emergency oil cylinder. See Fig. 36, arrangement of hydraulic operating mechanism
48 Spring
49 Spring
50 Collar for 49 and 52
51 Pin for 50 and 52
52 Trip rod
53 Bracket
54 Pin for 53
55 Spring
56 Trip finger
57 Pin and cotter for 53 and 56
58 Spiral gear casing
59 Minimum clearance between 56 and ring of emergency governor = $\frac{1}{4}$ in., maximum clearance $\frac{11}{32}$ in.
60 Position of emergency governor when thrown out



1 Ring
2 Bushing
3 Bushing for 5 and 7
4 Casing
5 Spring
6 Adjusting nut
7 Guide stud—riveted to 2
8 Rivet for 7
9 Washer for 5 and 6
10 Rivet for 1 and 4
11 Bolt for 2 and 14
12 Spacing stud
13 Guide pin—riveted to 2
14 Turbine shaft
15 Travel—$\frac{1}{8}$ in.
16 Bushing for 5 and 19
17 Adjusting nut
18 Washer for 5 and 17
19 Guide stud—riveted to 2
20 Machine screw for 17 and 18
21 Holes for pin spanner wrench

FIG. 12.—ASSEMBLY OF EMERGENCY GOVERNOR—DOUBLE RING TYPE.

responds to this imposed variation in load, the governor lever being lowered for increasing speed or raised for decreasing speed; this movement being continued until a balance is established on the floating levers and the pilot valve closes the admission ports to the hydraulic cylinder. The turbine will then continue to run at practically constant speed as long as the position of the speed lever is not altered.

Under certain conditions, such as in turning the ship, excessive load is thrown on the motors on the inboard side of the turning circle due to the action of the governor, which, if nothing prevented, would maintain a constant speed by admitting more steam to the turbine to compensate for the increased load. To prevent this action a steam limit device is installed, which when adjusted for any predetermined position, will allow only a definite number of controlling valves to be opened. This action is accomplished by a stop with attached indicating switch—sliding up or down with a connecting rod attached to the floating lever—which limits the travel of the pilot valve in one direction only. This stop with its indicating switch is controlled by the speed limit and speed control levers on the main switchboard in the control room and connected to the steam limit lever by an independent set of bell cranks and rods. While maneuvering, the steam limit lever must, of course, always be set so that a sufficiently wide margin of power be available for comparatively wide speed changes. However, when the ship is in free route and no more maneuvering bells are probable, the steam limit is then adjusted so that its stop is brought hard against the governor. Under this condition, of course any increase in load will cause a reduction in the turbine speed. Likewise with a decrease in load such as occurs in opening the field switch or in the propellers coming out of the water, the governor immediately shuts off the steam, thus preventing the turbine from racing. The operation of this device has been most satisfactory.

FIG. 12.—ASSEMBLY OF EMERGENCY GOVERNOR—DOUBLE RING TYPE.

LUBRICATING SYSTEM.

The main lubricating system for the turbine bearings, generators bearings, main motor bearings, main Kingsbury thrust bearings, and which also provides oil for the main hydraulic governor system, covers an arrangement of two distinct systems, one each on the port and starboard side. Pressure is supplied to either of these systems by two motor driven vertical pumps or, in case of the failure of these pumps, by a spiral gear oil pump driven directly from the governor end of each main turbine.

The motor driven pumps are rated at 300 gallons per minute at 1,500 R.P.M. to give 100 pounds pressure which normally will provide 75 pounds gauge pressure to the system after passing through various bafflers. Oil is supplied to the hydraulic cylinders at 75 to 80 pounds pressure and, by the use of bafflers and needle valves, is throttled down to from 5 to 10 pounds for the bearings.

MAIN CONDENSERS.

The *Maryland* is equipped with the conventional type of Navy, two-pass, surface condenser, having a total cooling surface of 12,050 square feet. The first or lower pass comprises 3,499 tubes and the upper or second pass 3,515 tubes, all the tubes being ⅝ inch outside diameter and 10 feet 8½ inches long. Of the tubes included in the lower pass 438 are in an air cooling space. The inside diameter of the shell is 9 feet 3 inches and the length between tube sheets 10 feet 6 inches. A single exhaust opening is provided on the top of the shell which is bolted to the lower side of the main turbine exhaust hood. A baffle plate is fitted over the air section to cool and condense the vapors. Entirely separate air ejection and condensate suctions are provided, the air being drawn from the uppermost part of the air cooling space located on the shell side opposite the exhaust opening.

Two separate systems for the removal of air and condensate are provided. The first is a twin beam, wet and dry air pump—Worthington-Blake-Knowles, Size 11 × 30 ×27. The second system and the one always in use except in case of breakdown, or for putting a daily vacuum on the condensers, consists of three steam air ejectors arranged in parallel together with a vertical motor driven condensate pump. The air ejectors are of the Westinghouse Le Blanc type, two-stage, size G, having the following rating:

Steam pressure, pounds per square inch	125
Pound of free air per hour at 28 inches vacuum referred to 30 inches barometer	36
Pounds of free air per hour at 28.5 inches vacuum referred to 30 inches barometer	27
Steam consumption, pounds per hour	880

The air ejectors discharge into air separating tanks or auxiliary condensers to condense the steam and to rid the condensate of air. The steam is condensed by the condensate pump discharge and hence the major part of the total heat content of the steam used in the ejectors is regained.

MAIN CONDENSATE PUMPS.

One motor driven double inlet, volute, vertical, single stage and single runner pump is provided for each condenser. The pump suction is taken off the main air pump suction and is discharged to the air separating tank. The complete unit is manufactured by the Westinghouse Electric Mfg. Co.

PUMP DATA.

Diameter suction pipe, inches	10
Diameter discharge pipe, inches	6
Capacity, gallons per minute	500
Head, feet	70
Motor, horsepower	19
voltage	240
speed, R.P.M.	1,400/1,700
windings, compound, interpole.	

MAIN CIRCULATING PUMP.

One Worthington centrifugal, horizontal, motor driven, variable speed, double inlet, single stage, main circulating pump is provided for each main condenser:

PUMP DATA.

Capacity, gallon per minute at 500 R.P.M.	19,000
Head, feet, at 500 R.P.M.	30
Diameter, suction, inches	28
discharge, inches	28
impeller, inches	25.5
Motor windings, shunt	
Motor, General Electric	
horsepower	125/250
voltage	115/230
speed, R.P.M.	400/500

MAIN GENERATORS.

The main generators are General Electric type AQB, Form HT, having a nominal rating of 13,400 K.V.A. at 2,065 R.P.M.; voltage, series connection of stator, 4,325; ampères 1,550; K.W. 11,000; power factor 82 per cent. Their maximum capacity is approximately 14,800 K.V.A.

The two phase windings of the stator are so arranged that they may be connected (1) in series or square for high voltage connection and (2) in parallel or diametrical for low voltage connection. The square or high voltage connection is used only when two generators are being used for main propulsion; that is, when the forward generator is connected to the two starboard main motors and the after generator to the port main motors. The diametrical or low voltage connection is used when only one generator is being used to propel the ship, the four main motors being tied in parallel. An interlock is provided so that the high voltage generator connection may not be used on one generator driving all four motors in order to prevent the overloading possible on one generator under such conditions.

The stator frame is constructed of a number of annular I-section iron castings secured together axially by steel supporting plates at the outer periphery and by steel punching ribs at their inner periphery. Heavy steel foot plates along each side of the stator frame are bolted to feet cast integrally as a part of the annular I-beams. Heavy galvanized iron lagging covers the stator frame serving to deaden the noise and also to direct the flow of ventilating air through and around the machine, the joints between lagging sections being covered by rolled brass beading strips.

The stator core, amply sectionalized for ventilating purposes, is built up of annealed, enameled thin sheet iron segments, with low iron loss characteristics. Heavy cast steel end flanges keyed to dove-tailed ribs hold the stator core under high compression thus eliminating vibrations in the punchings. The latter have open slots permitting the use of form wound coils.

The stator winding coils consist of two turns each, mica insulated, each turn consisting of many rectangular copper strips in multiple, each insulated from the other to prevent eddy current losses. Impregnated, specially treated wooden wedges secure the coils in the slots. The coil ends are securely laced to, and insulated from, steel binding bands to prevent coil distortion or coil insulation damage incident to heavy short circuits, or conditions approximating short circuits. Specially insulated terminal studs form connecting links between the stator windings and line cables, the beginning and end of each phase winding being brought out to a connection board. This board is mounted on brackets secured to the end I-beam section of the stator frame and is located at the collector end of the generator directly beneath the shaft. Three resistance temperature detectors are placed between the top and bottom coil sides and also between coils and the core at points of the core and winding most probably subject to the highest temperature. Spe-

cially protected leads from these detectors are brought to the main control room where they are connected to a temperature indicating instrument.

Sectional end shields are fitted at each end of the stator to deaden the generator noise, to guide the ventilating air entering the generator, and to prevent air pressure loss after leaving the fans.

MAIN GENERATOR ROTOR AND FIELD COILS.

The rotor or revolving field consists of a one-piece, heat-treated, solid steel forged shaft serving both as a field yoke and core. Radial slots, milled in the body portion of the shaft, carry the distributed field winding consisting of several coils per pole, there being only one pair of poles. Each rotor coil turn is formed from copper strip, wound edgewise, insulated with mica tape, and wound into the slot turn after turn. The complete coil is insulated from the rotor slot by a special combination armor and the coil is anchored in the slot by steel and brass wedges. Beyond the slots the coil end portions are insulated, blocked, and held in position by broad nickel steel retaining rings shrunk on to projecting lugs of the rotor body and over centering rings at the outer ends of the retaining rings. Holes are drilled in the retaining rings and a passage left between rings and rotor body for the necessary ventilation for the field coil ends.

The centering rings, besides supporting the retaining rings, also carry the multivane ventilating fans and whatever balancing weights are necessary.

The generator is further equipped with steam heating coils, located in the end shields at the bottom of the machine, to raise the temperature of the air inside the generator approximately five degrees above the room temperature and hence prevent accumulation of moisture on the stator and rotor windings. An additional small heating coil above the terminal board prevents the accumulation of moisture at the terminals.

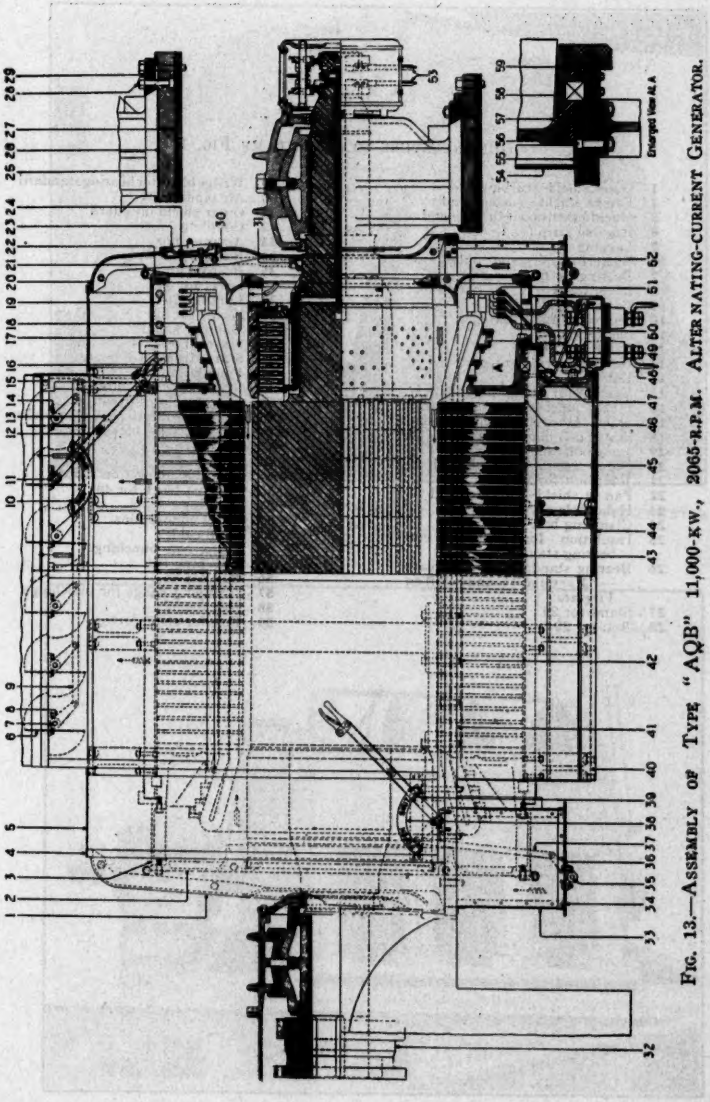


FIG. 13.—ASSEMBLY OF TYPE "AQB" 11,000-KW., 2065-R.P.M. ALTERNATING-CURRENT GENERATOR.

Nomenclature to Accompany Fig. 13.

1 Outer shield—turbine end
2 Fan fit shield—turbine end
3 Shield extension—turbine end
4 Beading strip for 1
5 Lagging
6 Butterfly damper air outlet
7 Bearing for 6
8 Lever for 6
9 Connecting rod for 6
10 Sector for outlet damper
11 Shaft for 6
12 Latch for 10
13 Handle for rotating part
14 Bolt for 7
15 Bracket for air discharge pipe
16 Stator punching clamping flange
17 Shield extension—collector end
18 Stator coil binding bands
19 Joint bolts for 17
20 Outer shield—collector end
21 Joint bolt for 22
22 Fan fit shield—collector end
23 Hand hole cover
24 Clamping bolt for 23
25 Insulation for collector end bearing standard
26 Bearing standard support plate —"*Maryland*" and "*West Virginia*"
27 Shims for 26
28 Bolt for 29
29 Wedge block for bearing standard
30 Air pipe for 31
31 Outer shield oil guard
32 Coupling
33 Air inlet box
34 Butterfly damper
35 Bearing for 34
36 Lever for 34
37 Link for 36
38 Sector for 39
39 Damper handle
40 Stator frame "I" beams
41 Stator frame foot plates
42 Foundation bolts
43 Stator core punchings
44 Support plates for 40
45 Inside ventilating space blocks
46 Outside space blocks
47 Air deflector
48 Line leads
49 Bus ring connection strips
50 Support block for 49
51 Bus rings
52 Series connections
53 Field cables
54 Stator core punchings
55 Punching rib
56 Bolt for 55
57 Clamping flange for punchings
58 Key for 57
59 Bolt for air deflector

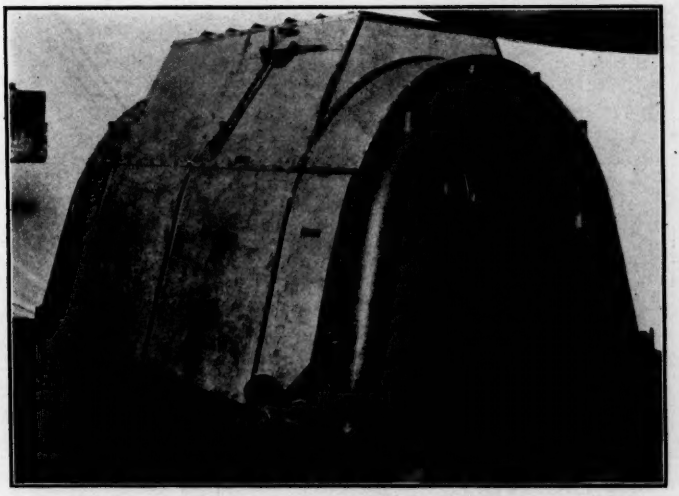


FIG. 14.—PROPULSION UNIT NO. 1 FOR U. S. S. "CALIFORNIA"—ALTERNATOR STATOR READY FOR SHIPMENT.

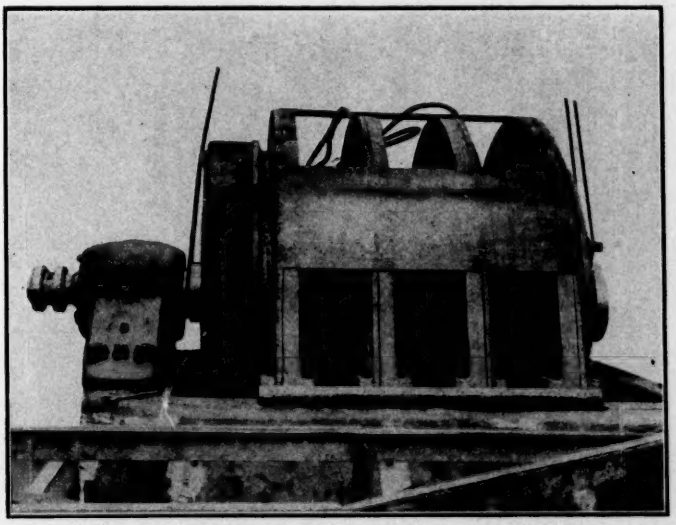


FIG. 15.—METHOD OF REMOVING ALTERNATOR ROTOR FROM STATOR BY MEANS OF ONE CRANE HOOK—POSITION NO. 1.

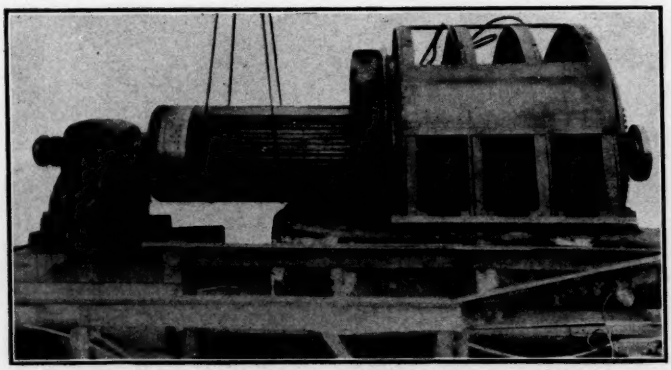


FIG. 16.—METHOD OF REMOVING ALTERNATOR ROTOR FROM STATOR BY MEANS OF ONE HOOK—POSITION NO. 5.

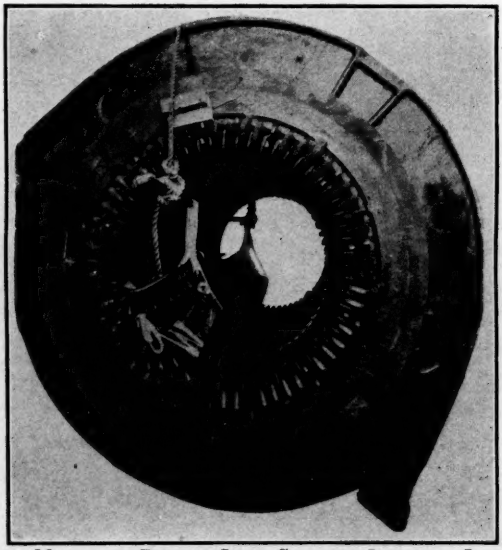


FIG. 17.—METHOD OF RAISING JUMP COILS FOR INSERTING LAST COIL.

FIG. 19.—TYPE 1Q-24/36-6700/1750-175/116 N 4200 V. INDUCTION MOTOR END SHIELD. SCREENS REMOVED. (COUPLING END.)

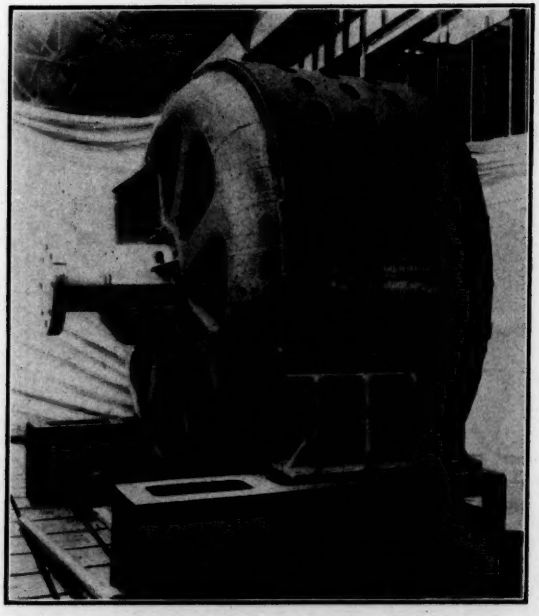


FIG. 20.—TYPE 1Q-24/36-6700/1750-175/116 N 4200 V. INDUCTION MOTOR END SHIELD. SCREENS REMOVED. (COUPLING END.)

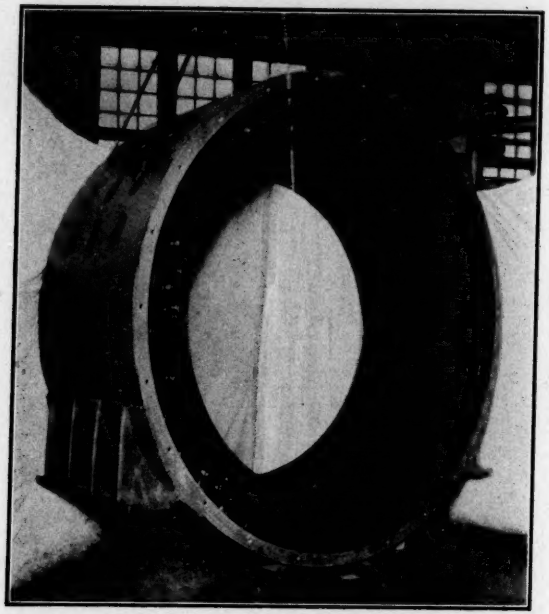


FIG. 21.—TYPE 1Q-24/36-7025/2125-170/118-N-4300/2480 V. INDUCTION MOTOR. STATOR LESS WINDING.

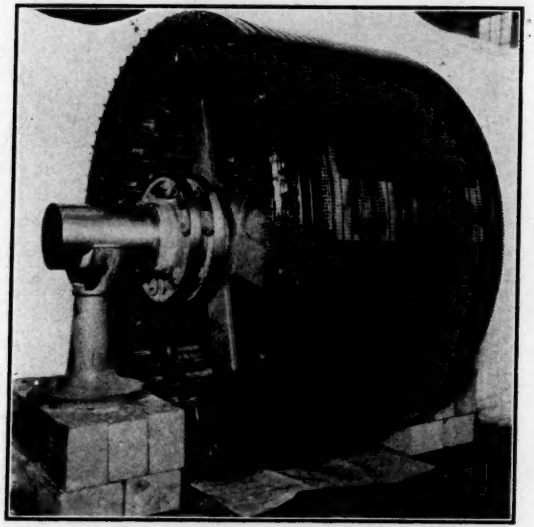


FIG. 22.—TYPE 1Q-24/36-7025/2125-170/118-N-4300/2480 V. INDUCTION MOTOR (ROTOR)

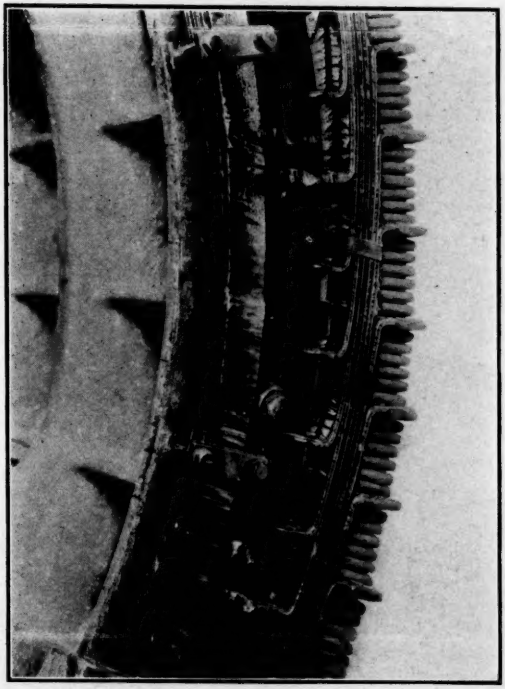


FIG. 23.—TYPE 1Q-24/36-6700/1750-175/116 N4200 INDUCTION MOTOR (SECTION OF ROTOR WINDINGS).

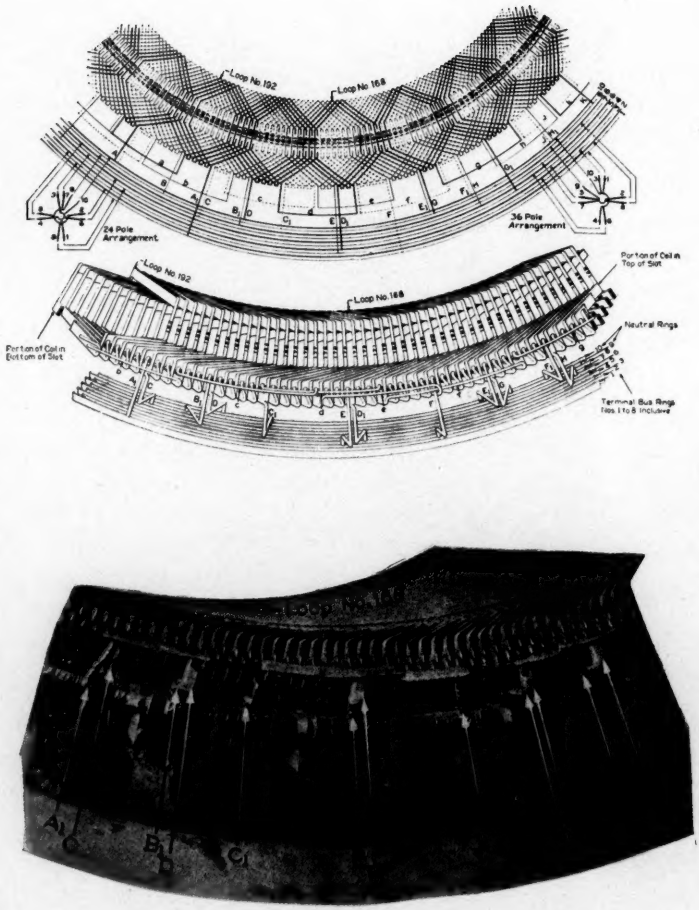

Loop No. 192
Loop No. 168
24 Pole Arrangement
36 Pole Arrangement
Portion of Coil in Top of Slot
Neutral Rings
Portion of Coil in Bottom of Slot
Terminal Bus Rings Taps 1 to 8 Inclusive

FIG. 24.—SECTION OF STATOR WINDING OF PROPULSION MOTORS TO ILLUSTRATE METHOD OF CUTTING OUT DEFECTIVE COILS.

Fire extinguishing is provided for by the installation of steam coils consisting of circular segments of brass tubing having a number of orifices drilled in them through which steam may be discharged over the windings, these segments being fastened to the fore and aft end shields at a position opposite the stator coils. Both inlet and outlet air dampers are fitted on the ventilation system and these dampers would be closed in case of fire.

The rotor is provided with bearings of the pedestal type, self aligning, fitted for forced lubrication. Provision is also made so that the air gap adjustment may be accurately made by shims. The pedestal on the collector end is insulated from the foundation to prevent circulating stray field currents damaging the bearings.

Main generator excitation is provided from any one of six 300 K.W. direct current, three wire, 120/240 volts turbo generating sets, there being three of these sets in each engine room, two being condensing sets and one non-condensing. The non-condensing sets are normally used to provide both the excitation power and that for the main propulsion motor driven auxiliaries. Two motor generator booster sets are also provided, one in each engine room, for giving wide variation to the excitation voltage.

Figures 13 to 17 illustrate details of main generator construction.

MAIN MOTORS

There are four two phase induction motors, each direct connected to one propeller shaft, the thrust of which is taken by a Kingsbury thrust bearing placed between the motor and propeller. These motors have a nominal rating of 7,000 S.H.P. each when running at a speed of 170 R.P.M. at an applied voltage of 4,400 with the generators square or series connected. The main motors are wound quarter phase with a primary winding in the stator so arranged that the number of

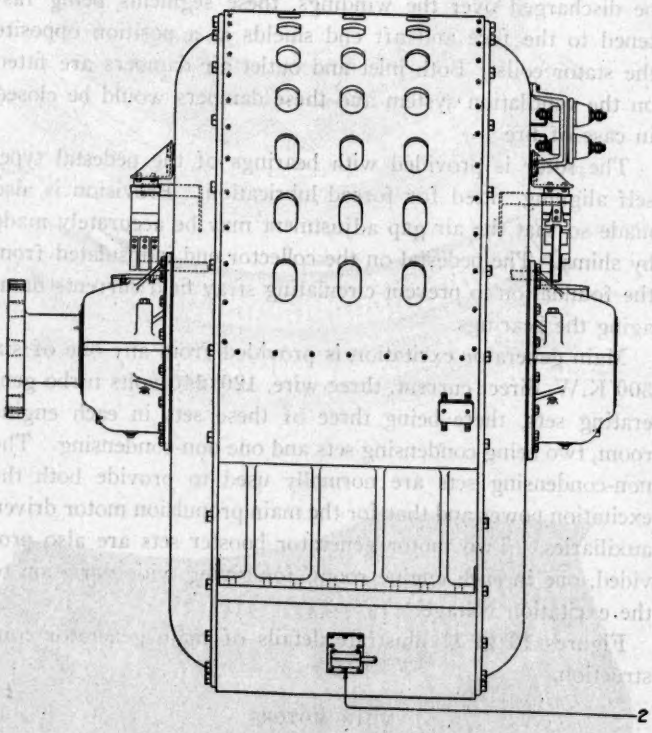


FIG. 18.—ASSEMBLY OF TYPE "IQ" INDUCTION MOTOR.

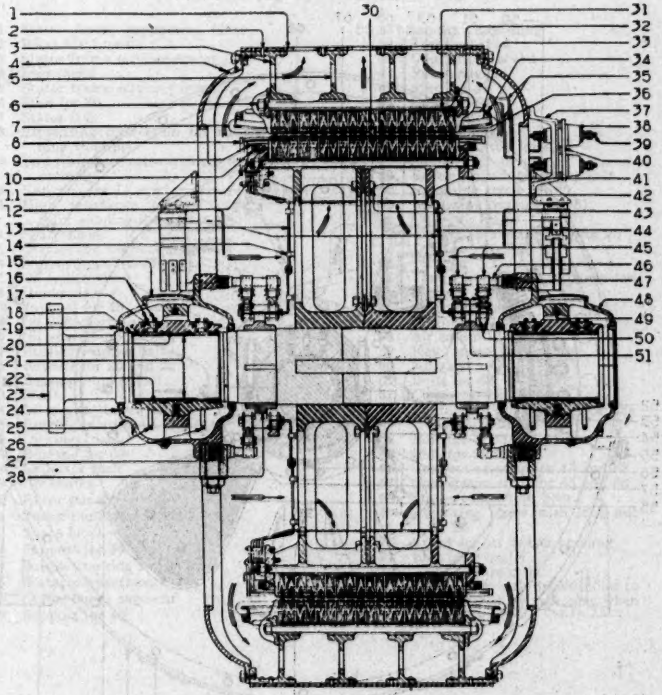


FIG. 18.—ASSEMBLY OF TYPE "IQ" INDUCTION MOTOR.

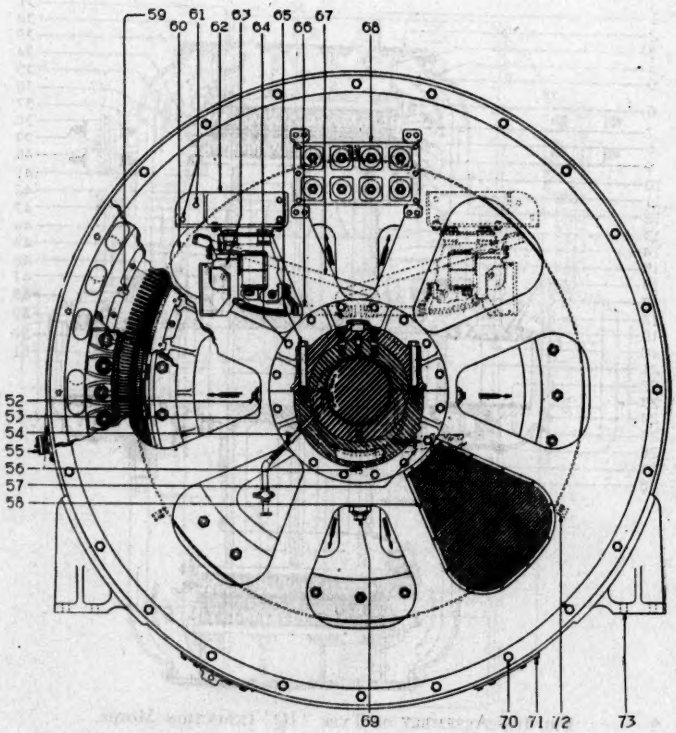


FIG. 18.—ASSEMBLY OF TYPE "IQ" INDUCTION MOTOR.

Nomenclature to Accompany Fig. 18.

1 Stator frame supporting plate
2 Rivet for 1 and 3
3 Stator frame section—outer
4 End shield
5 Stator frame section—inner
6 Stud for 36
7 Stator coil
8 Equalizing connection for running winding
9 Low resistance rotor winding for running
10 End ring for 11
11 High resistance squirrel cage rotor winding for starting
12 Space block
13 Connection between 9 and 51
14 Supporting cleat for 13
15 Oil well cover
16 Guide pins
17 Support for 16
18 Oil deflectors on shaft
19 Oil guard—upper half
20 Bearing—upper half
21 Babbitt for 20 and 22
22 Bearing—lower half
23 Shaft
24 Oil guard—lower half
25 Bearing housing—lower half
26 Oil rings
27 Bushing for 69
28 Collector shell
29 Air heater
30 Rotor punching
31 Stator punching
32 Space block
33 Support for 34
34 Supporting ring for 7
35 Stator connections
36 Stator flange segment
37 Support for 40
38 Insulating bushing
39 Terminal stud
40 Connection board
41 Rotor flange
42 Stud for 41
43 Bolt for 44
44 Rotor spider
45 Brush-holders
46 Insulated stud for 45
47 Support for 46
48 Bearing housing—upper half
49 Retaining screw for 20
50 Insulating bushing for 51
51 Collector rings
52 Lateral adjusting screw for 25
53 Lock nut for 52
54 Oil inlet
55 Temperature coil connection board
56 Oil drain plug
57 Oil outlet
58 Air screen
59 Key for stator punching
60 Air gap
61 Bolts for 62
62 Contactor support
63 Contactor for short-circuiting rotor windings
64 Bolt for 4, 25 and 48
65 Bolt for 25 and 48
66 Connection strip for 45 and 63
67 Connection strip for 45 and 63
68 Stator connection board
69 Adjusting screw with lock nut for 25
70 Cover for air heater opening
71 Bolt for 3 and 4
72 Temperature coils
73 Motor foot with drilled holes to be reamed by customer when installing

poles can be changed from 24 to 36. Since the generator has but two poles the revolutions per minute of the motor will approximate respectively 1/12 or 1/18 of the R.P.M. of the generator. Up to and including 15 knots, therefore, or 118 R.P.M. of the main motors, the 36-pole connection is used—corresponding to a turbine speed of 2,170 R.P.M.; all intermediate speed changes being made by the turbine governor. Beyond 15 knots to maximum speed, the 24-pole connection of the motor stators is used. For instance, at 15 knots the motor speed of 118 R.P.M.—on the 24-pole connection—corresponds to a turbine speed of 1,440 R.P.M. All intermediate changes up to maximum speed are made through the turbine governor. Thus, within the propeller speed range is included two points at which the turbine is running at maximum speed and hence best economical condition, a feature largely accounting for the high efficiency of electric propulsion. Furthermore, one generator alone is used up to 17 knots, the motor pole connections being changed at 15 knots from 36 to 24. The experience of the *Maryland* has further shown that in an emergency one generator can assume a greater load than that for 17 knots, a feature which might well become paramount in action.

The stator frames of the main motors consists of cast steel I-beam section rings held rigidly in position by riveting to solid steel castings forming the feet, which in turn are bolted to foundations forming part of the ship's structure. The ring sections are united by boiler plates rolled to shape and riveted. The core is built up of segmental laminations stamped from thin steel, each segment being independently held by slots to dove-tailed ribs secured to the I-beam sections and both sides of the laminations are enameled to prevent eddy current losses. Radial air ducts are provided at frequent intervals along the length of the core opposite to and in line with similar ducts in the rotor to provide for ventilation through the core and around the windings. These ducts are maintained by means of steel fingers of I-beam section spot welded to one adjacent lamina-

tion, the laminations being clamped between stiff end plates with fingers for supporting teeth and held in position by heavy bolts.

The stator winding consists of wound coils of rectangular wire insulated with mica tape and with an external layer of tape treated with special moisture resisting varnish. The completed coils are firmly wedged in the slots of the stator punchings, the external coil ends being laced to steel rings supported from the stator frame to prevent vibration.

The rotor is built up of two cast steel spiders keyed to the shaft and bolted to each other, while the rotor core is built of thin sheet steel segmental laminations, each segment of which is independently held by ribs fitting in dove-tailed slots of the rotor spider. Radial ducts as described for the stator are also provided in the rotor for ventilation.

The rotor has two independent windings in the same slots, an outer low resistance, high inductance phase winding for continuous running and an inner high resistance, low inductance squirrel-cage winding for starting. The outer winding nearest the air gap is of copper heavily insulated with mica. To each phase of this winding is connected a pair of slip rings, one pair at each end of the rotor. The brush leads from these collector rings are led to a contactor mounted on the motor end shield by means of which the winding can be opened or closed on itself. The contactors are energized from a switch operated by the main generator field switch lever in the control room. With a 36-pole stator connection and contactors open through non-closure of the contactor switch controlled by the main field lever the phase windings are opened and inoperative, but with the same stator pole connection and the contactor switch closed by the main field lever, the phase windings are closed and operative, they being the high power factor running windings of the rotor. However, with a 24-pole stator connection the phase winding is closed on itself without the necessity of closing the contactors, the full load current not passing through collector rings or brushes.

In the bottom of the rotor slots are placed the inner high resistance low inductance German silver bars of the squirrel-cage windings, their ends brazed to short circuiting end rings. These latter bars are separated from the outer copper windings by thick insulating strips. This inner squirrel-cage winding is active only during starting and reversing operations with the motor connected for 36 poles and the contactors open. Once the motor is pulled into step by the squirrel-cage winding—its prime function—the outer phase-wound copper windings are thrown into circuit by action of contactors energized through main generator field lever as previously described.

The outer frame sections of the motor stator are extended to form supports for the motor end-shields, each end-shield consisting of a solid steel casting to which are bolted the bearing housings. The bolt holes for these housings have clearance to allow for air gap adjustment by means of three adjusting screws serving as screw jacks for raising or lowering the rotors. The bearing housings themselves support self aligning bearings, the upper half being removable for examination and a lubricating system of the oil ring type is fitted.

Ventilation is furnished each motor by two 10,000 cubic feet blowers mounted above the motor, drawing air through openings in the end-shields thence through the rotor and stator axial and radial air ducts and delivering it finally to large exhaust ducts which carry it up and out of the ship. For preventing moisture precipitation when the motor is at rest, each motor is fitted with an adjustable 500-2,000 watt electric heating unit installed in the motor stator frame. To indicate temperature conditions in the rotor stator three temperature coils are imbedded in the stator slots 120 mechanical degrees apart around the periphery, their ends being brought out and connected to the temperature control board in the main control room where the indications of each coil are recorded through suitable switching connections to a double scale temperature indicator.

The motor construction is shown in Figures 18 to 24, inclusive.

MAIN PROPULSION AUXILIARIES.

The following auxiliary equipment necessary for the operation of the main generators and motors is installed:

2–300 K.W. non-condensing geared turbo generator sets.
4–300 K.W. condensing geared turbo generator sets.
2–19,000 G.P.M. motor driven main circulating pumps.
2–1,000 G.P.M. motor driven dynamo circulating pumps.
4–80/250 G.P.M. motor driven lubricating oil pumps.
8–10,000 cubic feet per minute motor driven main motor blowers.
2–Oil cooler circulating pumps, motor driven.
2–18 K.W.-60 volt motor generator boosters.

The above pumps are all motor driven and controlled from the D.C. engine room switchboards by General Electric automatic control panels of the CR Type. The ventilating blower motors are controlled by switches on the main motor ventilation control panel in the main control room through hand operated control panels of the CR type at the motor itself. The booster motor is started by means of a pump type switch mounted on the D.C. engine room switchboard. The control wiring for these units is shown in Figures 25 to 29, inclusive.

The following is the data on the main propulsion auxiliaries; the main circulating pumps, lubricating oil pumps and main motor blowers having heretofore been described.

300 K.W. TURBO GENERATOR DATA.

Turbine (Condensing):

Stages	3
Form	E
Type	HT
Steam pressure	200 pounds
R.P.M.	5037

Generator:

Form	O
Type	MPC
K.W.	300
Volts	120/240
Ampères	1250
R.P.M.	1000
Shunt current	9

Reduction Gear:

Type	SR
R.P.M.	5037/1000

The non-condensing turbine differs only from the condensing type in that it is a single stage turbine, Form A-HT, operating at a steam pressure of 250 pounds.

DYNAMO CIRCULATING PUMP DATA.

Pump:

Type—Cameron, Form 3DS, Volute, Horizontal.

Capacity	2200 G.P.M.
Head	20 feet
R.P.M.	1100/1425
B.H.P.	18
Size, impeller	9 inches
Size, suction	10 inches
Size, discharge	9 inches

Motor:

Type—G.E. Co.; RA11B, Form A1.

Ampères	69
Volts	230
H.P.	18
R.P.M.	1100/1425
Winding	Compound

LUBRICATING OIL PUMP DATA.

Pump:

Type—G.E. Co.; rotary, single stage.
Capacity ..... 300 G.P.M.
R.P.M. 1000/1500

Motor:

Type—G.E. Co.; RC-32, Form A, Compound.
Ampères 70/130
Volts 115/230
H.P. 16/34
R.P.M. 1000/1500

OIL COOLER CIRCULATING PUMP DATA.

Pump:

Type—Cameron, double suction, Volute.
Capacity ..... 300 G.P.M.
Head ..... 115 feet
R.P.M. 1800/2200

Motor:

Type—G.E. Co.; Form A1, Type RC11, Compound.
Ampères 58.5
Volts 230
H.P. 15.4
R.P.M. 1800/2200

BOOSTER GENERATOR SET DATA.

Motor:

Type—G.E. Co.; RC12, Form A, Shunt wound.
Ampères 105
Volts 240
H.P. 30
R.P.M. 1000/1600

Generator:

Type—G.E. Co.; RC12, Form A, Shunt wound.
Ampères 300
Volts 60
K.W. 18
R.P.M. 1000/1600

MAIN CONTROL EQUIPMENT.

The main control equipment consists of the following:

(*a*) Two D.C. engineroom switchboards each of which consists of five panels, one each for each 300 K.W. generator and two for controlling the motor generator booster and the ten engineroom auxiliary motors together with the main motor ventilation supply. (Figures 26 and 27.)

(*b*) One motor ventilation control panel supplying the main propulsion blower motors and the constant voltage D.C. control circuit for energizing all magnetic locks, contactors, and indicating lamps. (Figures 30 and 31.)

(*c*) One alternating current switchboard consisting of a structural steel cell with equipment for controlling both main generators and the four main motors.

D. C. ENGINE ROOM SWITCHBOARDS.

Each D.C. engine room switchboard controls three 300 K.W. 120/240, three wire, D.C. generators; one 18 K.W. 60-Volt, booster and ten engine room auxiliaries. These switchboards are divided into five panels as indicated above and shown in Figure 26. Each generator circuit is fitted with one four-pole automatic, time limit, overload, and reverse current circuit breaker. Two triple-pole lever switches are also fitted in the generator circuit by means of which the generator may be connected either to the bus feeding the engine room auxiliaries and excitation for the A.C. generators or the power and light bus or both. One of these triple-pole switches is made double throw so that when in port either the power and light bus or

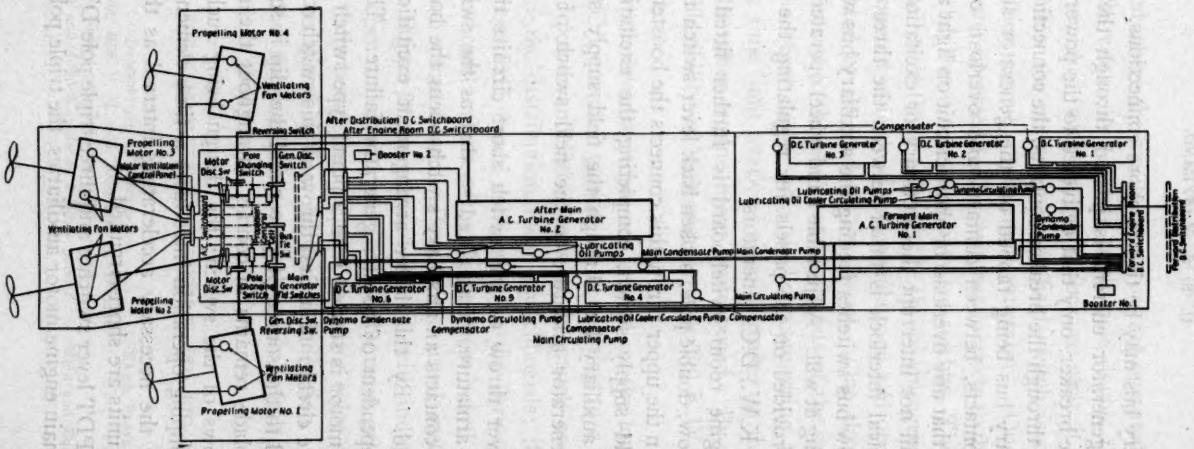


FIG. 25.—DIAGRAMMATIC ARRANGEMENT OF PROPULSION CONTROL EQUIPMENT.

the auxiliary bus may be tied to shore connections. Although the total generator current is passed through the overload coils of the breaker, only that delivered to the power and light bus passes through the breaker contacts, the connection feeding the auxiliary bus being tapped off the generator side of the breaker contacts, between contacts and overload coil. This is done so that any overload or short circuit on light and power circuits will not interrupt the auxiliary or excitation circuits. A mechanical interlock is fitted between the three triple-pole single throw bus switches feeding the auxiliary bus with means for releasing at will. Adjustable hand wheel operated rheostats are also provided on the panels for regulating the excitation of the 300 K.W., D.C. generators.

The engine room switchboard is further fitted with one double throw, double pole, broken back lever switch for booster control. In the upper throw this connects the booster armature to the field supply switch, completing the excitation circuit from the auxiliary bus through the field supply switch and booster generator armature to the field switch of the A.C. generator.

The lower throw of the switch short circuits the booster generator armature, being fitted so that as the switch leaves the upper contacts an auxiliary switch opens the booster generator field. By this latter arrangement excitation can be made independent of booster generator failure. The booster generator motor is started with a pump type switch located in the positive circuit. The first throw of this switch connects the shunt field to the source of power and the clip is so arranged that the blade remains in contact throughout the succeeding three throws of the switch; the design being such that the handle pumping operation introduces a time element allowing the motor the necessary accelerating interval as the various resistance units are short-circuited.

Nine DPDT lever switches and three triple pole DT switches feed the main engine motor auxiliaries, the triple pole switches

being fitted where the auxiliary motors are operated for either 240 or 120 volts. A separate triple pole double throw lever switch, furnished with magnetic lock so that it cannot be opened unless the main generator field switch is open, feeds the excitation circuit of the main generators. One throw of all these lever operated feeder switches connects to the auxiliary bus, the other throw to an auxiliary reserve bus. This latter bus is connected to a triple pole double throw emergency transfer switch mounted in back of the switchboard and operated by a lever located at the end of the board. One throw of this switch connects to the power and light bus of the same switchboard; the other throw to a tie line connecting through an automatic time limit overload circuit breaker to the auxiliary bus of the other engine room switchboard. By this means complete flexibility of both engine room 300 K.W. generators is secured.

Single pole lever switches are located on the lower sections of the D.C. generator panels for connecting to a tie line between the power and light buses on the panels and the buses of one of the distribution room switchboards; the forward engine room D.C. board connecting to the forward distribution room board and the after engine room board to the after distribution room board. Besides the usual instruments as shown in Figure 27, each board is fitted with indicating lamps to show open and closed positions of the main generator field breaker, there being two lamps of the same color connected in multiple to guard against failure of indication. Two other lamps below the ventilation control feeder switch indicate that this circuit is alive.

FORWARD D.C. ENGINE ROOM BOARD, FIG. 26.

Part No.	Description of Apparatus.	Name Plate Inscriptions.
1	CK-106, 250 V. 4 pole 1250/500 A. Inverse time limit overload circuit breaker with shunt trip and divided studs on 2 inner poles.....	Main and Equalizer.

1-A	RB-14YN Reverse current Relay for Part No. 1	
2	D-9 Ammeter 2000 A, with shunt	Negative.
3	D-9 Voltmeter 300 V	Generator No. 1.
4	D-9 Ammeter 2000 A. with shunt	Positive
5	D-9 Voltmeter 300 V	Generator No. 2.
6	D-9 Voltmeter 300 V	Generator No. 3.
7	D-127, 250 V. 1000/2000 A. TPST lever switch	Ship service — Light and Power.
8	Rheostat Handwheel and Mechanism	Field Rheostat.
9	D-127, 250 V. 1000/2000 A. TPST lever switch	Main drive auxiliaries.
11	Sangamo Swbd. type DS Watt hour-meter 3 wire, 120/240 V. 2000 A. with shunt	Watt hour meter.
12	D-127, 250 V. 600 A. SPST lever switch	Forward Distribution Swbd.
13	D-127, 250 V. 3000 A. SPST lever switch	Forward Distribution Swbd.
14	D-127, 250 V. 3000 A. SPST lever switch	Forward Distribution Swbd.
15	D-9 Ammeter 500 A. with shunt	A.C. Generator Field.
16	Ruby Bull's eye indicating Lamp	A.C. Generator Field in.
17	Green Bull's eye indicating Lamp	A.C. Generator Field out.
18	D-9 Ammeter 200-0-200 A. with shunt	Booster Motor.
19	D-9 Ammeter 2000 A. scale with 1500 A. shunt	Total Ampères Forward.
20	D-9 Ammeter 2000 A. scale with 1500 A. shunt	Total Ampères Aft.
21	D-9 Voltmeter 350 V	A.C. Generator Field.
22	CP-106, 250 V. 200 A. DP. circuit breaker with inverse time limit overload coil and undervoltage release and circuit opening auxiliary switch	Booster Motor.
23	Ships nameplate	*Maryland.*
24	D-9 Voltmeter 300 V	Aft Auxiliary Bus.
25	D-9 Ammeter 2000 A. scale with 1500 A. shunt	Total Ampères For'd.
26	CK-106, 250 V. 3 pole 1250/500 A. with time limit overload on outside 1250 A. poles	Aft Auxiliary Bus.
27	D-9 Ammeter 2000 A. scale with 1500 A. shunt	Total Ampères Aft.
28	D-9 Ammeter 1200 A. with shunt	Main Circulating Pump.
29	D-9 Voltmeter 300-0-300 V	Grounds.
30	11 Point single pole ground switch	For'd Main Bus For'd Aux, Bus, Aft Aux, Bus. Blank—Grd.—Grd., Grd—, Grd.,—Grd., Grd—, Grd., Grd., Grd—, Blank.
31	Bureau Nameplate	
32	D-9 Voltmeter 300 V	Calibrating.
33	6 point DP Voltmeter switch	—Voltage—Gen. No. 1, Gen. No. 2, Gen. No. 3, Booster, Aft Bus, Blank.
34	D-0 Ammeter 100 A. with shunt	Dynamo Circulating Pump.
35	D-9 Ammeter 150 A. with shunt	Oil Cooler Pump.
36	D-9 Ammeter, 150 A. with shunt	Forced Lubrication Pump No. 1.
37	D-9 Ammeter, 80 A. with shunt	Dynamo Hot Well Pump.
38	D-9 Ammeter, 150 A. with shunt	Forced Lubrication Pump No. 2.
39	D-9 Ammeter, 150 A. with shunt	Main Hot Well Pump.
40	D-127, 250 V. 300 A. TPDT lever switch with magnetic lock	Booster For'd.
41	D-127, 250 V. 300 A. DPDT lever switch with single pole circuit opening auxiliary switch	Booster Generator in.

42	D-127, 250 V. 300 A. TPDT lever switch with Ruby bull's eye ind. lamp	Main to Motor Ventilating Blowers—For'd.
43	D-127, 250 V. 150 A. DPDT lever switch	Oil Cooler Pump—For'd.
44	D-127, 250 V. 150 A. TPDT lever switch	Forced Lubrication Pump No. 1—For'd.
45	D-127, 250 V. 150 A. TPDT lever switch	Forced Lubrication Pump No. 2—For'd.
46	D-127, 250 V. 150 A. DPDT lever switch	Main Hot Well Pump—For'd.
47	D-127, 250 V. 150 A. TPDT lever switch with magnetic lock	Booster—Aft.
48	D-127, 250 V. 300 A. TPDT lever switch with single pole circuit opening auxiliary switch	Booster Generator Disconnected.
49	D-127, 250 V. 300 A. TPDT lever switch with Ruby bull's eye indicating lamp	Main to Motor Ventilating Blower—Aft.
50	D-127, 250 V. 150 A. DPDT lever switch	Oil Cooler Pump—Aft.
51	D-127, 250 V. 150 A. TPDT lever switch	Forced Lubrication Pump No. 1—Aft.
52	D-127, 250 V. 150 A. TPDT lever switch	Forced Lubrication Pump No. 2—Aft.
53	D-127, 250 V. 150 A. DPDT lever switch	Main Hot Well Pump—Aft.
54	D-127, 250 V. 1000 A. TPDT lever switch	Main Circulating Pump—For'd.
55	D-127, 250 V. 1000 A. TPDT lever switch	Main Circulating Pump—Aft.
56	H-16, 250 V. 150 A. SPDT starting switch	Booster Motor.
57	D-127, 250 V. 150 A. DPDT lever switch	Dynamo Circulating Pump—For'd.
58	D-127, 250 V. 75 A. DPDT lever switch	Ventilating Blower, No. 1—For'd.
59	D-127, 250 V. 75 A. DPDT lever switch	Ventilating Blower No. 2—For'd.
60	D-127, 250 V. 75 A. DPDT lever switch	Spare (no marking)
61	D-127, 250 V. 75 A. DPDT lever switch	Dynamo Hot Well Pump—For'd.
62	D-127, 250 V. 75 A. DPDT lever switch	Ventilating Blower No. 3—For'd.
63	D-127, 250 V. 75 A. DPDT lever switch	Ventilating Blower No. 4—For'd.
64	D-127, 250 V. 75 A. DPDT lever switch	Dynamo Circulating Pump—Aft.
65	D-127, 250 V. 75 A. DPDT lever switch	Ventilating Blower No. 1—Aft.
66	D-127, 250 V. 75 A. DPDT lever switch	Ventilating Blower No. 2—Aft.
67	D-127, 250 V. 75 A. DPDT lever switch	Dynamo Hot Well Pump—Aft.
68	D-127, 250 V. 75 A. DPDT lever switch	Ventilating Blower No. 3—Aft.
69	D-127, 250 V. 75 A. DPDT lever switch	Ventilating Blower No. 4—Aft.
70	Potential fuse	Undervolt Release Booster Circuit Breaker.
71	Potential fuse	For'd Aux. Bus Volts.
72	30 A. 250 V. NECS Encl. fuse	A.C. Generator Field.

73	30 A. 250 V. NECS Encl. fuse	General Illumination.
74	30 A. 250 V. NECS Encl. fuse	Booster Generator Field.
75	Potential fuse	For'd Aux. Bus Volts.
76	Potential fuse	Power and Light Bus Volts.
77	Potential fuse	Spare (no markings).
78	150 & 600 A. 250 V. NECS Encl. fuse	Main to Motor Ventilating Blower.
79	150 & 160 A. 250 V. NECS Encl. fuse	Spare Fuses—Main to Motor Vent. Blowers.
80	Potential and 30 A. 250 V. NECS Encl. fuse	Generator No. 1.
81	Potential and 30 A. 250 V. NECS Encl. fuse	Generator No. 2.
82	Potential and 30 A. 250 V. NECS Encl. fuse	Generator No. 3.
83	Potential fuse	Rev. Cur. Pot. Coil.
84	Potential fuse	Voltmeter.
85	30 A. 250 V. NECS. Encl. fuse	Shunt Trip.

AFTER D.C. ENGINE ROOM BOARD, FIG. 27.

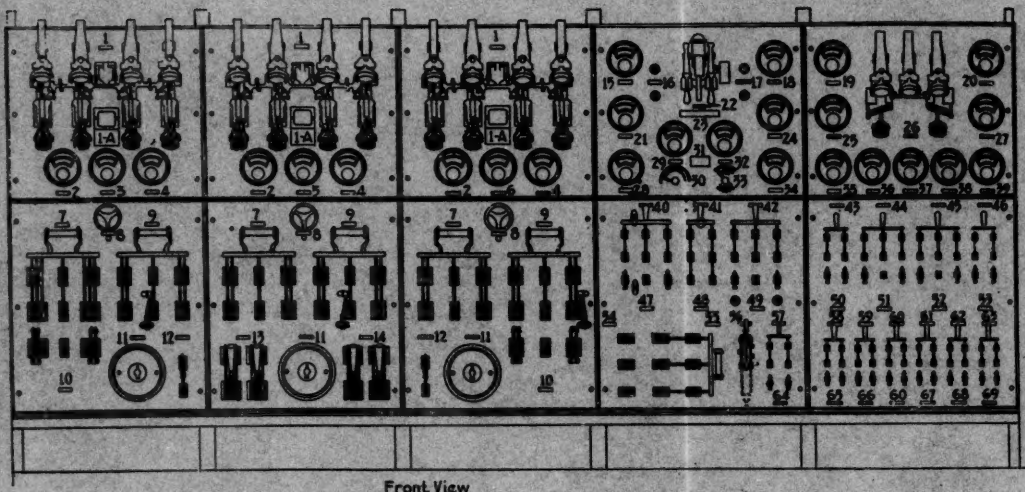
Part No.	Description of Apparatus.	Name Plate Inscriptions.
1	CK-106, 250 V. 4 pole 1250/500 A. Inverse time limit overload circuit breaker with shunt trip and divided studs on 2 inner poles	Main and Equalizer.
1-A	RB-14YN Reverse Current Relay for Part No. 1	
2	D-9 Ammeter 2000 A. with shunt	Negative.
3	D-9 Voltmeter 300 V	Generator No. 4.
4	D-9 Ammeter 2000 A. with shunt	Positive
5	D-9 Voltmeter 300 V	Generator No. 5
6	D-9 Voltmeter 300 V	Generator No. 6.
7	D-127, 250 V. 1000/2000 A. TPDT (and TPST) lever switch	Ships Service — Light and Power.
8	Rheostat Handwheel and Mechanism	Field Rheostat.
9	D-127, 250 V. 1000/2000 A. TPDT (and TPST) lever switch	Main Drive Auxiliaries.
10	D-127, 250 V. 1000/2000 A. TPDT lever switch	Shore Connections.
11	Sangamo Swbd. Type DS Watt hour-meter, 3 wire 120/240 V. 2000 A. with shunt	Watt Hour Meter.
12	D-127, 250 V. 600 A. SPST lever switch	After Distribution Swbd.
13	D-127, 250 V. 3000 A. SPST lever switch	After Distribution Swbd.
14	D-127, 250 V. 3000 A. SPST lever switch	After Distribution Swbd.
15	D-9 Ammeter 500 A. with shunt	A.C. Generator Field.
16	Ruby Bull's eye indicating lamp	A.C. Generator Field in.
17	Green Bull's Eye indicating lamp	A.C. Generator Field Out.
18	D-9 Ammeter 200-0-200 A. with shunt	Booster Motor.
19	D-9 Ammeter 2000 A. scale with 1500 A. shunt	Total Ampères Aft.
20	D-9 Ammeter 2000 A. scale with 1500 A. shunt	Total Ampères For'd.
21	D-9 Voltmeter 350 V	A.C. Generator Field.
22	CP-106, 250 V. 200 A. D.P. circuit breaker with inverse time limit overload coil and undervoltage release and circuit opening auxiliary switch	Booster Motor.
23	Ship's nameplate	*Maryland.*
24	D-9 Voltmeter 300 V	For'd Auxiliary Bus.

25	D-9 Ammeter 2000 A. scale with 1500 A. shunt	Total Ampères Aft.
26	CK-106, 250 V. 3 pole 1250/500 A. with time limit overload on outside 1250 A. poles	For'd Auxiliary Bus.
27	D-9 Ammeter 2000 A. scale with 1500 A shunt	Total Ampères For'd.
28	D-9 Ammeter 1200 A. with shunt	Main Circulating Pump.
29	D-9 Voltmeter 300-0-300 V	Grounds.
30	11 Point single pole ground switch	Aft Main Bus, Aft Aux. Bus, For'd Aux. Bus. Blank—Grd—Grd., Grd.—Grd.,—Grd., Grd.—, —Grd.,—Grd., Grd.—, Blank.
31	Bureau nameplate	
32	D-9 Voltmeter 300 V	Calibrating.
33	6 Point D.P. Voltmeter switch	—Voltage— Gen. No. 4, Gen. No. 5, Gen. No. 6, Booster, Fo'rd Bus, Blank.
34	D-9 Ammeter 100 A. with shunt	Dynamo Circulating Pump.
35	D-9 Ammeter 150 A. with shunt	Oil Cooler Pump.
36	D-9 Ammeter 150-A. with shunt	Forced Lubrication Pump No. 1.
37	D-9 Ammeter 80 A. with shunt	Dynamo Hot Well Pump.
38	D-9 Ammeter 150 A. with shunt	Forced Lubrication Pump No. 2.
39	D-9 Ammeter 150 A. with shunt	Main Hot Well Pump.
40	D-127, 250 V. 300 A. TPDT lever switch with magnetic lock	Booster—For'd.
41	D-127, 250 V. 300 A. DPDT lever switch with single pole circuit opening auxiliary switch	Booster Generator In.
42	D-127, 250 V. 300 A. TPDT lever switch with Ruby Bull's Eye indicating lamp	Main to Motor Ventilating Blowers—Aft.
43	D-127, 250 V. 150 A. DPDT lever switch	Oil Cooler Pump—Aft.
44	D-127, 250 V. 150 A. TPDT lever switch	Forced Lubrication Pump No. 1—Aft.
45	D-127, 250 V. 150 A. TPDT lever switch	Forced Lubrication Pump No. 2—Aft.
46	D-127, 250 V. 150 A. DPDT lever switch	Main Hot Well Pump—For'd.
47	D-127, 250 V. 300 A. TPDT lever switch with magnetic lock	Booster—For'd.
48	D-127, 250 V. 300 A. DPDT lever switch with single pole circuit opening auxiliary switch	Booster Generator Disconnected.
49	D-127, 250 V. 300 A. TPDT lever switch with Ruby Bull's Eye indicating lamp	Main to Motor Ventilating Blower—For'd.
50	D-127, 250 V. 150 A. DPDT lever switch	Oil Cooler Pump—For'd.
51	D-127, 250 V. 150 A. TPDT lever switch	Forced Lubrication Pump No. 1—For'd.
52	D-127, 250 V. 150 A. TPDT lever switch	Forced Lubrication Pump No. 2—For'd.
53	D-127, 250 V. 150 A. DPDT lever switch	Main Hot Well Pump—For'd.
54	D-127, 250 V. 1000 A. TPDT lever switch	Main Circulating Pump—Aft.
55	D-127, 250 V. 1000 A. TPDT lever switch	Main Circulating Pump—For'd.
56	H-16, 250 V. 150 A. SPDT starting switch	Booster Motor.

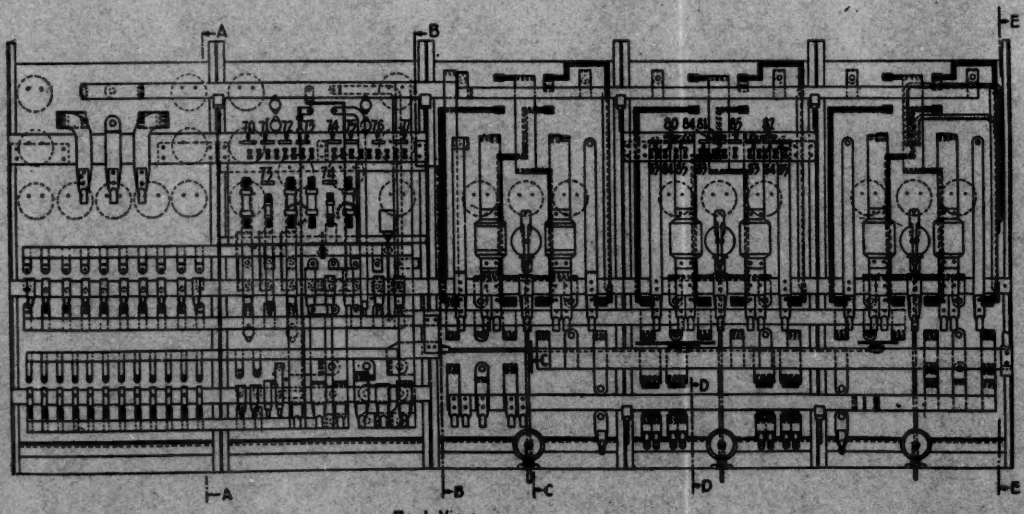
57	D-127, 250 V. 150 A. DPDT lever switch	Dynamo Circulating Pump—Aft.
58	D-127, 250 V. 75 A. DPDT lever switch	Ventilating Blower No. 1—Aft.
59	D-127, 250 V. 75 A. DPDT lever switch	Ventilating Blower No. 2—Aft.
60	D-127, 250 V. 75 A. DPDT lever switch	Spare (no marking).
61	D-127, 250 V. 75 A. DPDT lever switch	Dynamo Hot Well Pump—Aft.
62	D-127, 250 V. 75 A. DPDT lever switch	Ventilating Blower No. 3—Aft.
63	D-127, 250 V. 75 A. DPDT lever switch	Ventilating Blower No. 4—Aft.
64	D-127, 250 V. 75 A. DPDT lever switch	Dynamo Circulating Pump—For'd.
65	D-127, 250 V. 75 A. DPDT lever switch	Ventilating Blower No. 1—For'd.
66	D-127, 250 V. 75 A. DPDT lever switch	Ventilating Blower No. 2—For'd.
67	D-127, 250 V. 75 A. DPDT lever switch	Dynamo Hot Well Pump—For'd.
68	D-127, 250 V. 75 A. DPDT lever switch	Ventilating Blower No. 3—For'd.
69	D-127, 250 V. 75 A. DPDT lever switch	Ventilating Blower No. 4—For'd.
70	Potential fuse	Undervolt, Release, Booster Circuit Breaker.
71	Potential fuse	For'd Aux. Bus Volts.
72	30 A. 250 V. NECS Encl. fuse	A.C. Generator Field.
73	30 A. 250 V. NECS Encl. fuse	General Illumination.
74	30 A. 250 V. NECS Encl. fuse	Booster Generator Field.
75	Potential fuse	Aft Aux. Bus Volts.
76	Potential fuse	Power and Light Bus Volts.
77	Potential fuse	Spare (no markings).
78	150 & 600 A. 250 V. NECS Encl. fuse	Main to Motor Ventilating Blower.
79	150 & 600 A. 250 V. NECS Encl. fuse	Spare Fuses—Main to Motor Vent. Blowers.
80	Potential and 30 A. 250 V. NECS Encl. fuse	Generator No. 4.
81	Potential and 30 A. 250 V. NECS Encl. fuse	Generator No. 5.
82	Potential and 30 A. 250 V. NECS Encl. fuse	Generator No. 6.
83	Potential fuse	Rev. Cur. Pot. Coil.
84	Potential fuse	Voltmeter.
85	30 A. 250 V. NECS Encl. fuse	Shunt Trip.

MOTOR VENTILATION AND TEMPERATURE INDICATING PANEL.

This panel, located in the control room, is fed by lines from each engine room switchboard. On it are mounted on the lower section eight double pole double throw switches feeding all motor driven blowers for ventilating main motors. The top of this panel is fitted with a triple pole double throw transfer switch for energizing all control circuits together

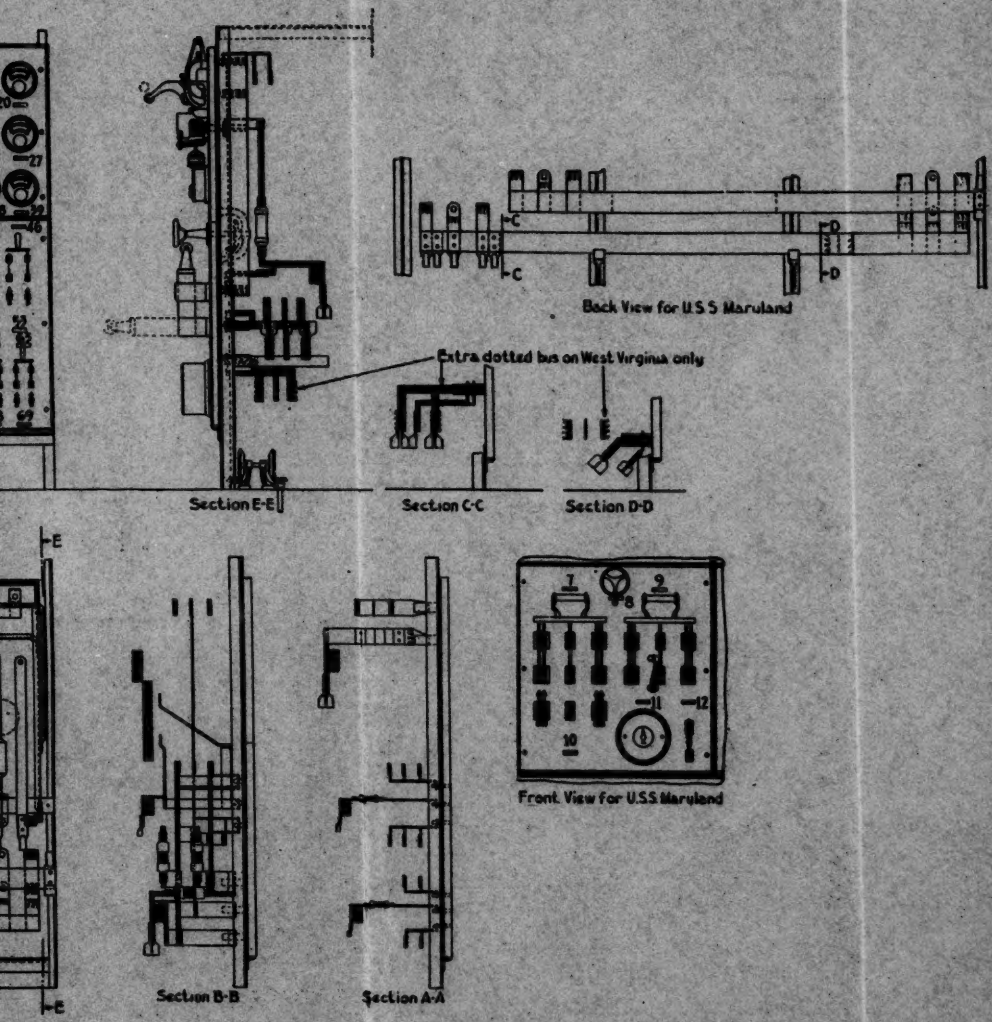


Front View

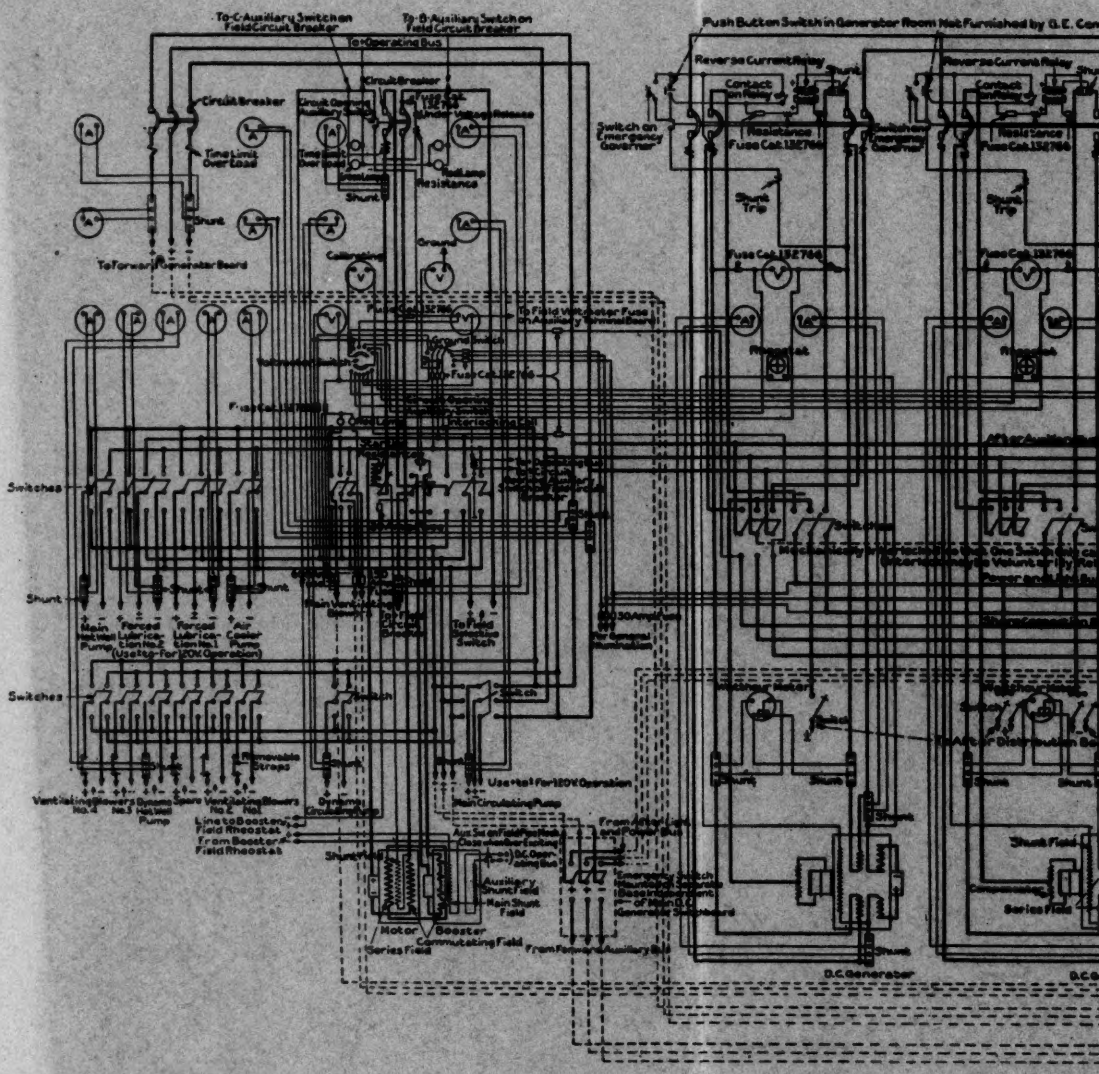


Back View

FIG. 27.—AFTER D-C. ENGINE ROOM SWITCHBOAR



M SWITCHBOARD—U. S. S. "MARYLAND."


After Board
To C Auxiliary Switch on Field Circuit Breaker
To D Auxiliary Switch on Field Circuit Breaker
To Operating Bus
Push Button Switch in Generator Room Not Furnished by G.E. Co
Circuit Breaker
Reverse Current Relay
Contactor
Resistance
Shunt Trip
Time Limit Over Load
Shunt
Ballast Resistance
Ground
To Forward Generator Board
Switches
Main Motor Pump
Forced Lubrication Pump
Forced Lubrication Motor
Air Cooler Motor
(Use for 120V Operation)
Main Circulating Pump
Use for 120V Operation
Ventilating Blowers No.4
Ventilating Blowers No.3
Dynamic Brake Motor
Ventilating Blowers No.2
Ventilating Blowers No.1
Line to Booster Field Rheostat
From Booster Field Rheostat
Motor Series Field
Booster Commutating Field
Auxiliary Shunt Field
Main Shunt Field
From Forward Auxiliary Bus
D.C. Generator
Shunt Field
Series Field

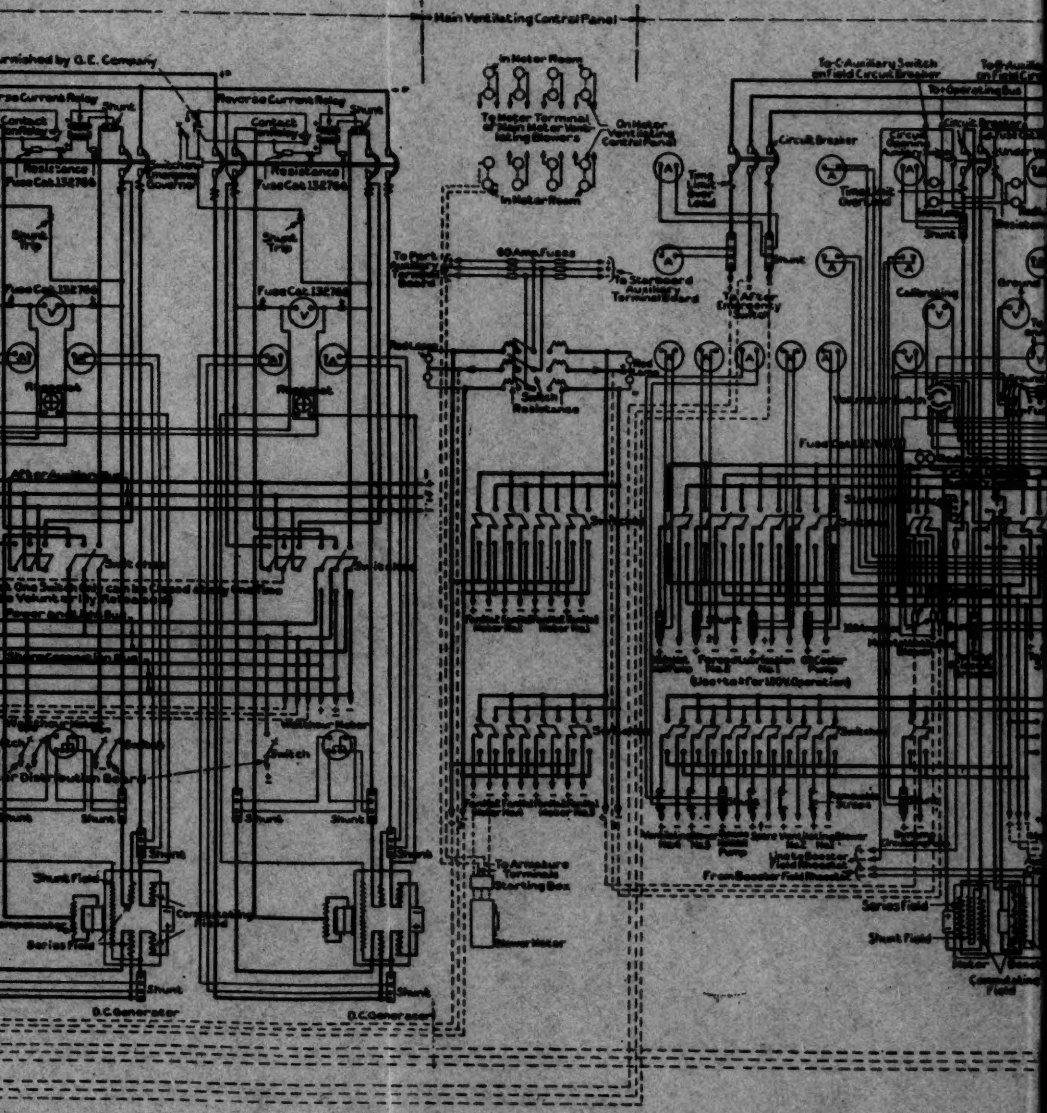


FIG. 28.—WIRING DIAGRAM, D-C. ENGINE ROOM SWITCHBOARD.

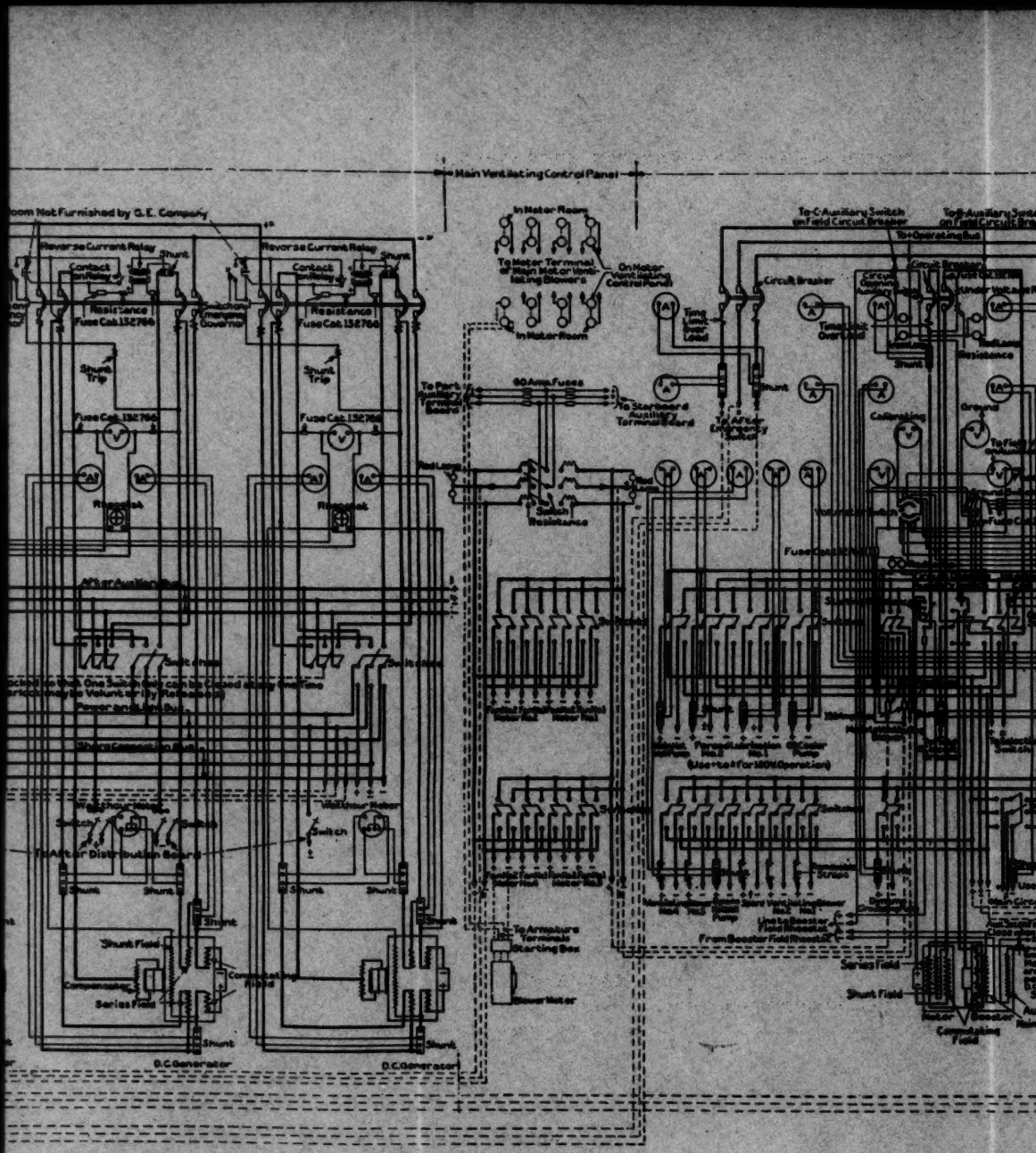
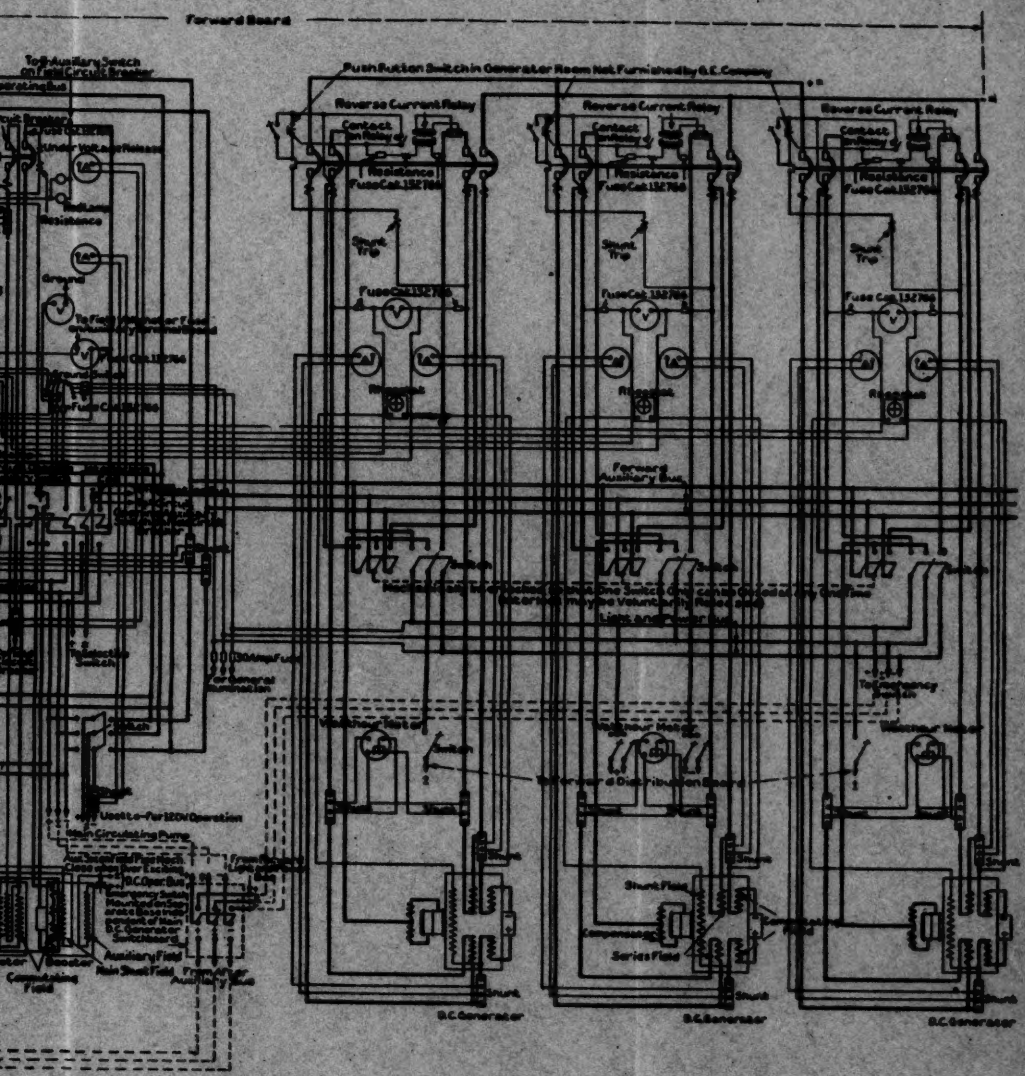
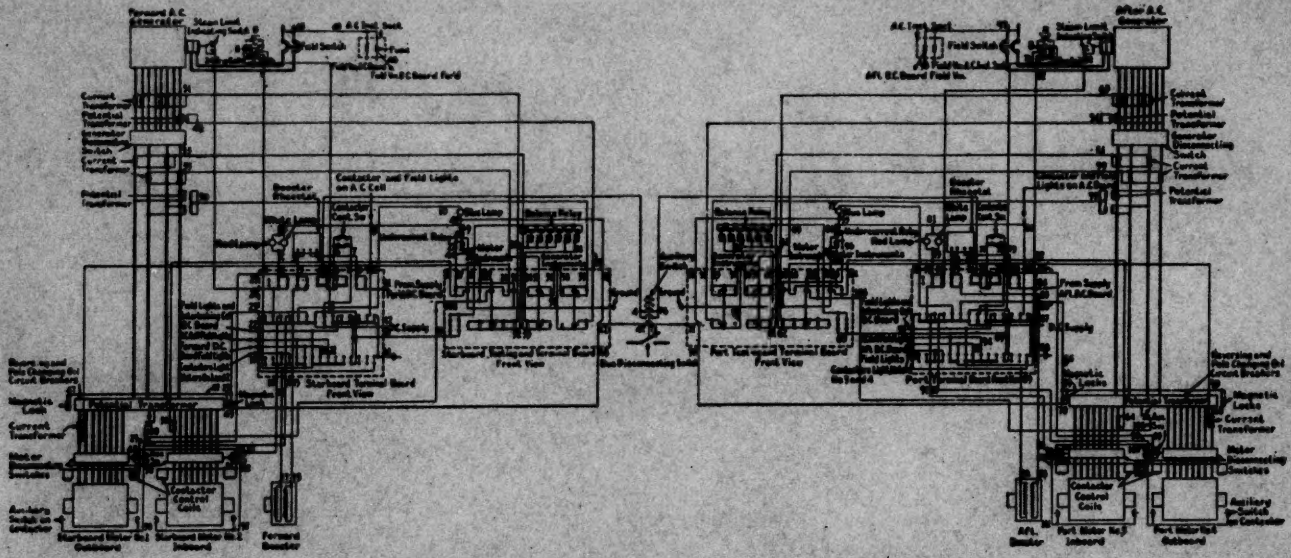
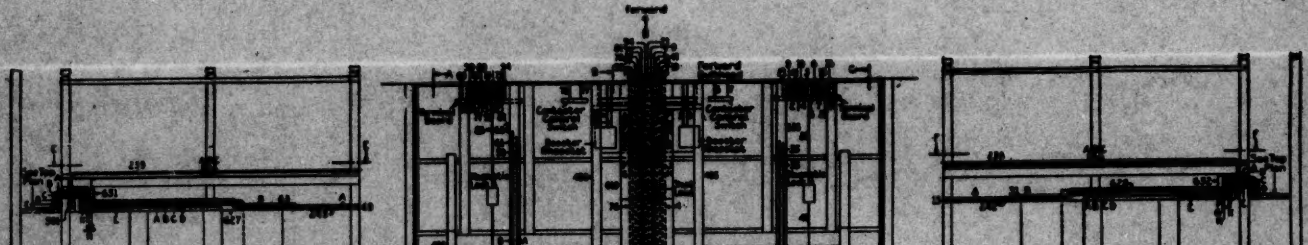


FIG. 28.—WIRING DIAGRAM, D-C. ENGINE ROOM SWITCHBOARD.


Forward Board
Push Button Switch in Generator Room Not Furnished by G.E. Company
Reverse Current Relay
Contact Auxiliary
Resistance
Fuse Cat. 152766
Shunt Trip
Ground
Reactance
Forward Auxiliary Bus
Main Circulating Pump
Auxiliary Field
Main Shunt Field
Series Field
Shunt Field
Compensating Field
D.C. Generator



Wiring Diagram of Cable Runs—U. S. S. *Maryland* and *West Virginia*



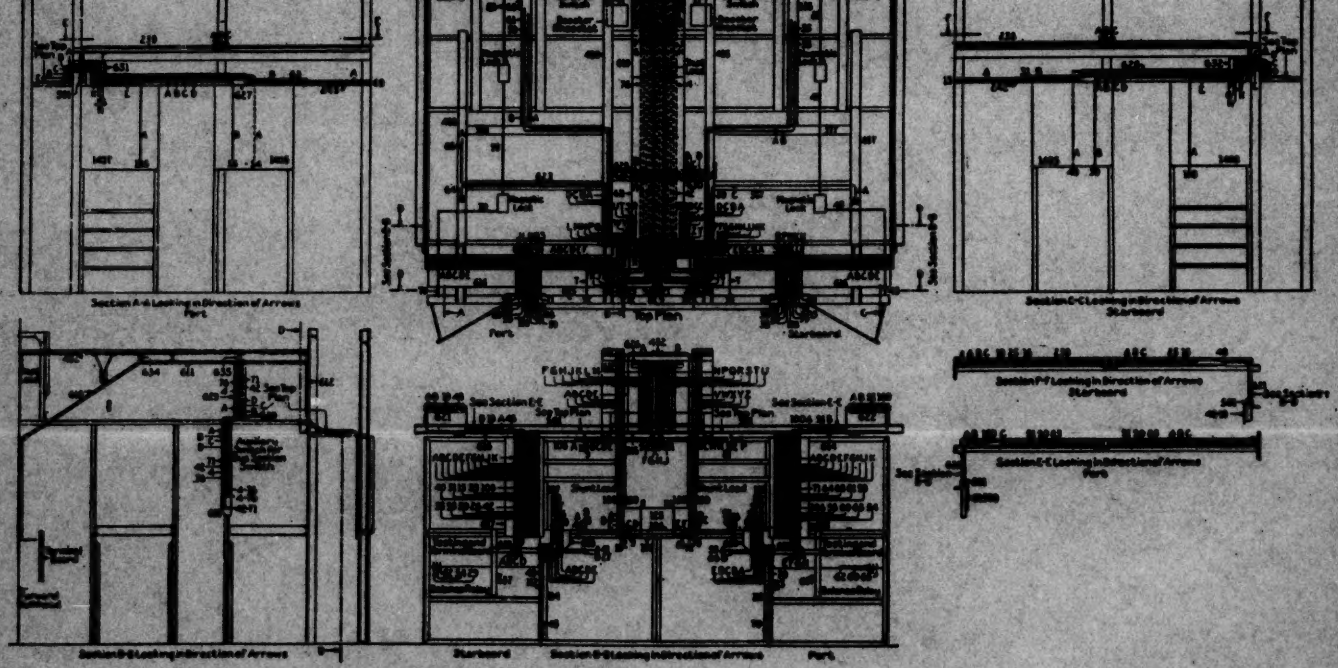


FIG. 29.—ARRANGEMENT OF CABLE RUNS—U. S. S. "MARYLAND."

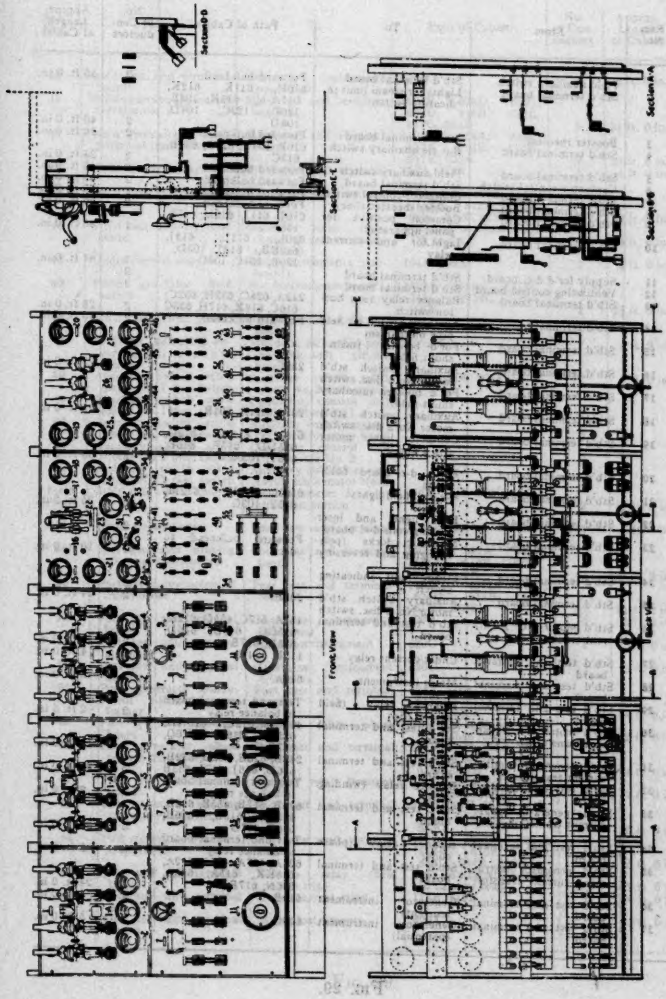


FIG. 26.—FORWARD D-C. ENGINE ROOM SWITCHBOARD—U. S. S. "MARYLAND."

Run No.	From	To	Path of Cables	No. of Conductors	Approx. Length of Cables
1	Field auxiliary switch	Stb'd terminal board	Forward bulkhead	2	15 ft. 0 in.
2	Stb'd terminal board	Lights for steam limit indicating switch	610K, 611K, 612K, 615A-A, 615K, 103E, 120C, 123C, 104D, 106D	2	40 ft. 0 in.
3	Booster rheostat	Stb'd terminal board	Forward bulkhead	2	16 ft. 0 in.
4	Stb'd terminal board	Bus tie auxiliary switch	610P, 611P, 629B, 630B, 613C	2	34 ft. 0 in.
5	Stb'd terminal board	Field auxiliary switch	Forward bulkhead	2	15 ft. 0 in.
6	Contactor control switch	Stb'd terminal board	Forward bulkhead	2	15 ft. 0 in.
7	Stb'd terminal board	Contactor control switch	Forward bulkhead	2	15 ft. 0 in.
8	Stb'd terminal board	Booster rheostat (line)	Forward bulkhead	2	16 ft. 0 in.
9	Stb'd terminal board	Common positive for panel apparatus	610J, 611J, 612L, 615Z, 103F	1	35 ft. 0 in.
10	Stb'd terminal board	Light for under-current relay	610L, 611L, 612J, 615BB, 615J, 103D, 120B, 104C, 106C	1	44 ft. 0 in.
11	Supply for'd d-c. board	Stb'd terminal board		2	
12	Ventilating control board	Stb'd terminal board		3	
13	Stb'd terminal board	Balance relay turn button switch	242A, 626C, 632H, 632C, 616C, 616K, 617H, 633C	2	28 ft. 0 in.
14	Stb'd terminal board	Auxiliary switch on field pipe mechanism	Forward bulkhead	2	15 ft. 0 in.
15	Stb'd terminal board	For'd booster (main shunt field)		2	
16	Stb'd terminal board	Auxiliary switch stb'd motor No. 1 disc. switch	238C	2	23 ft. 0 in.
17	Stb'd terminal board	For'd booster (auxiliary shunt field)		2	
18	Stb'd terminal board	Auxiliary switch stb'd motor No. 2 disc. switch	238A, 621A, 501B	2	25 ft. 0 in.
19	Stb'd terminal board	Contactor lights motor No. 1 and 2	610N, 611N, 612G, 615DD, 615P, 616F, 616T, 103A	2	36 ft. 0 in.
20	Stb'd terminal board	For'd d-c. board field lights		2	
21	Stb'd terminal board	A-c. cell field lights	610H, 611H, 612M, 615Y, 103G	2	35 ft. 0 in.
22	Stb'd terminal board	Field lights and interlocking coils d-c. board		2	
23	Stb'd terminal board	Magnetic locks (pole changing and reversing switches)	Forward bulkhead to deck	1	10 ft. 0 in.
24	Stb'd terminal board	Steam limit indicating switch		3	
25	Stb'd terminal board	Auxiliary switch stb'd motor No. 2 disc. switch	238B	2	24 ft. 0 in.
26	Stb'd motor No. 1 current transformer	Stb'd test and terminal board	487A, 517C, 455D, 612D, 615GG, 615D, 616D, 616R, 617B	2	20 ft. 0 in.
27	Stb'd test and terminal board	Under-current relay	112B, 114B	2	4 ft. 0 in.
28	Stb'd test and terminal board	Motor instrument	619A	2	5 ft. 0 in.
29	Stb'd test and terminal board	Balance relay (field switch trip)	Test and terminal board to balance relay	2	2 ft. 0 in.
30	Stb'd motor No. 2 current transformer	Stb'd test and terminal board	455C, 612C, 615HH, 615C, 616C, 616Q, 617C	2	15 ft. 0 in.
31	Stb'd generator winding current transformer	Stb'd test and terminal board	242B, 626B, 632G, 632B, 616D, 616J, 617J, 633B	5	20 ft. 0 in.
32	Stb'd test and terminal board	Balance relay (winding coils)	Test and terminal board to balance relay	5	2 ft. 0 in.
33	Stb'd generator phase current transformer	Stb'd test and terminal board	625B, 517B, 455B, 612B, 615JJ, 615B, 616B, 616P, 617D	3	24 ft. 0 in.
34	Stb'd test and terminal board	Balance relay (phase coils)	Test and terminal board to balance relay	3	2 ft. 0 in.
35	Stb'd generator instrument current transformer	Stb'd test and terminal board	625A, 517A, 455A, 612A, 615KK, 615A, 616A, 616N, 617E	3	24 ft. 0 in.
36	Stb'd test and terminal board	Generator instrument (current)	619B	3	5 ft. 0 in.
37	Stb'd test and terminal board	Generator instrument (potential)	619C	2	5 ft. 0 in.

FIG. 29.

Run No.	From	To	Path of Cables	No. of Conductors	Approx. Length of Cables
38	Stb'd test and terminal board	Generator instrument (potential)	619D	3	3 ft. 0 in.
39	Stb'd generator potential transformer	Stb'd test and terminal board	1405B, 628D, 632J, 632D, 616B, 616L, 617G, 633D	3	21 ft. 0 in.
40	Stb'd generator winding potential transformer	Stb'd test and terminal board	1405A, 628A, 632P, 632A, 616E, 616H, 617K, 633A	2	21 ft. 0 in.
41	Under-current relay	Stb'd test and terminal board	112A, 114A	1	4 ft. 0 in.
42	Stb'd test and terminal board	Bus tie auxiliary switch	617A, 616S, 616E, 615E, 615EE, 612F, 611P, 629C, 630C, 613B	1	33 ft. 0 in.
43	Stb'd test and terminal board	Magnetic locks (pole changing and reversing switches)	Test and terminal board 112 to deck	1	10 ft. 0 in.
44	Under-current relay	Motor instruments (star-board)	104A, 106A	2	2 ft. 0 in.
45	For'd generator field switch	A-c. instruments panel	610M, 611M, 612H, 615C, 615H, 120A, 104B	2	25 ft. 0 in.
46	Stb'd instrument panel	Field voltmeter for'd d-c. board		1	
47	Magnetic lock stb'd reversing oil circuit breaker	Magnetic lock stb'd pole changing coil circuit breaker	On deck	2	3 ft. 0 in.
48	Auxiliary switch stb'd motor No. 1 disc. switch	Auxiliary switch stb'd motor No. 2 disc. switch	238B, 621B, 501A	2	7 ft. 0 in.
49	Auxiliary switch stb'd motor No. 1 disc. switch	Contactor control coil stb'd motor No. 1		2	
50	Auxiliary switch stb'd motor No. 1 disc. switch	Auxiliary switch on contactor stb'd motor No. 1		2	
51	Auxiliary switch stb'd motor No. 2 disc. switch	Contactor control coil stb'd motor No. 2		2	
52	Auxiliary switch stb'd motor No. 2 disc. switch	Auxiliary switch on contactor stb'd motor No. 2		2	
Field shunt lead	Stb'd field switch shunt	Field ammeter a-c. instrument section	452A, 612E, 615Q, 615G, 103B	2	27 ft. 0 in.
53	Under-current relay	Port test and terminal board	113A, 115A	1	4 ft. 0 in.
54	Port generator winding potential transformer	Port test and terminal board	1405A, 627A, 631F, 631A, 614E, 614H, 618K, 633H	2	21 ft. 0 in.
55	Port generator potential transformer	Port test and terminal board	1405B, 627D, 631J, 631D, 614B, 614L, 618G, 633E	3	21 ft. 0 in.
56	Port test and terminal board	Generator instrument (potential	620B	3	5 ft. 0 in.
57	Port test and terminal board	Generator instrument	620C	2	5 ft. 0 in.
58	Port test and terminal board	Generator instrument (current)	620D	3	5 ft. 0 in.
59	Port generator instrument current transformer	Port test and terminal board	624A, 518A, 456A, 612Z, 615L, 615A, 614A, 614N, 618E	3	34 ft. 0 in.
60	Port test and terminal board	Balance relay phase coils	Test and terminal board to balance relay	3	2 ft. 0 in.
61	Port generator phase current transformer	Port test and terminal board	624B, 518B, 456B, 612Y, 615M, 615B, 614B, 614P, 618D	3	24 ft. 0 in.
62	Port test and terminal board	Balance relay (winding coils)	Test and terminal board to balance relay	5	2 ft. 0 in.
63	Port generator winding current transformer	Port test and terminal board	243B, 627B, 631G, 631B, 614D, 614J, 618J, 633G	5	30 ft. 0 in.
64	Port motor No. 3 current transformer	Port test and terminal board	458B, 623B, 456D, 612W, 615P, 615D, 614D, 614R, 618B	2	15 ft. 0 in.
65	Port test and terminal board	Balance relay (field switch trip)	Test and terminal board to balance relay	2	2 ft. 0 in.
66	Port test and terminal board	Under-current relay	113B, 115B	2	4 ft. 0 in.
67	Port test and terminal board	Motor instruments	620A	2	5 ft. 0 in.

FIG. 29.

Run No.	From	To	Path of Cables	No. of Conductors	Approx. Length of Cables
68	Port motor No. 4 current transformer	Port test and terminal board	458A, 623A, 456C, 612X, 615N, 615C, 614Q, 618C	2	20 ft. 0 in.
69	Port terminal board	Balance relay turn button switch	243A, 627C, 631H, 631C, 614C, 614K, 618H, 633F	2	28 ft. 0 in.
70	Port test and terminal board	Magnetic locks (pole changing and reversing switches)	Test and terminal board 113 to deck	1	10 ft. 0 in.
71	Port test and terminal board	Bus tie auxiliary switch	618A, 614S, 614E, 615E, 615R, 612U, 611A, 629D, 630D, 613A	1	33 ft. 0 in.
72	Port terminal board	Light for under-current relay	610D, 611D, 612R, 615U, 615J, 103K, 106E, 120D, 123D, 104E	1	41 ft. 0 in.
73	Port terminal board	Lights for steam limit indicating switch	610E, 611E, 612Q, 615V, 615K, 103L, 120E, 123E, 104F, 106F	2	40 ft. 0 in.
74	Port terminal board	Steam limit indicating switch		3	
75	Booster rheostat	Port terminal board	Forward bulkhead	2	16 ft. 0 in.
76	Port terminal board	Bus tie auxiliary switch	610A, 611A, 629A, 630A, 613D	2	34 ft. 0 in.
77	Contactor control switch	Stb'd terminal board	Forward bulkhead	2	15 ft. 0 in.
78	Port terminal board	Contactor control switch	Forward bulkhead	2	15 ft. 0 in.
79	Port terminal board	Field auxiliary switch	Forward bulkhead	2	15 ft. 0 in.
80	Port terminal board	Booster rheostat (line)	Forward bulkhead	2	16 ft. 0 in.
81	Port terminal board	Common positive for panel apparatus	610F, 611F, 612P, 615W, 103J	1	35 ft. 0 in.
82	Field auxiliary switch	Port terminal board	Forward bulkhead	2	15 ft. 0 in.
83	Port terminal board	Auxiliary switch port motor No. 3 disc. switch	230C	2	24 ft. 0 in.
84	Supply aft d-c. board	Port terminal board		2	
85	Port terminal board	Aft booster (main shunt field)		2	
86	Port terminal board	Magnetic lock (pole changing and reversing switches)	Forward bulkhead to deck	1	10 ft. 0 in
87	Ventilating control board	Port terminal board		3	
88	Port terminal board	Auxiliary switch field lever	Forward bulkhead	2	15 ft. 0 in.
89	Port terminal board	Aft booster auxiliary shunt field		2	
90	Port terminal board	Auxiliary switch port motor No. 4 disc. switch	239B	2	23 ft. 0 in.
91	Port terminal board	Auxiliary switch port motor No. 3 disc. switch	239A, 622A, 501B	2	25 ft. 0 in.
92	Port terminal board	Contactor lights motor No. 3 and 4	610B, 611B, 612T, 615S, 615F, 614F, 614T, 103F	2	36 ft. 0 in.
93	Port terminal board	Aft d-c. board field lights		2	
94	Port terminal board	A-c. cell field lights	610G, 611G, 612N, 615X, 103H	2	35 ft. 0 in.
95	Port terminal board	Field lights and interlocking coil on aft d-c. board		2	
96	Under-current relay	Motor instruments (port)	104H, 106H	2	2 ft. 0 in.
97	Aft generator field switch	A-c. instrument panel	610C, 611C, 612S, 615T, 615H, 103M, 120F, 104G	2	25 ft. 0 in.
98	Port instrument panel	Field voltmeter on aft d-c. board		1	
99	Magnetic lock port reversing oil circuit breaker	Magnetic lock (pole changing oil circuit breaker)	Deck	2	3 ft. 0 in.
100	Auxiliary switch port motor No. 3 disc. switch	Auxiliary switch port motor No. 4 disc. switch	239B, 622B, 501A	2	7 ft. 0 in.
101	Auxiliary switch port motor No. 3 disc. switch	Auxiliary switch on contactor port motor No. 2		2	
102	Auxiliary switch port motor No. 3 disc. switch	Contactor control coil port motor No. 3		2	
103	Auxiliary switch port motor No. 4 disc. switch	Auxiliary switch on contactor port motor No. 4		2	
104	Auxiliary switch port motor No. 4 disc. switch	Contactor control coil port motor No. 4		2	

FIG. 29.

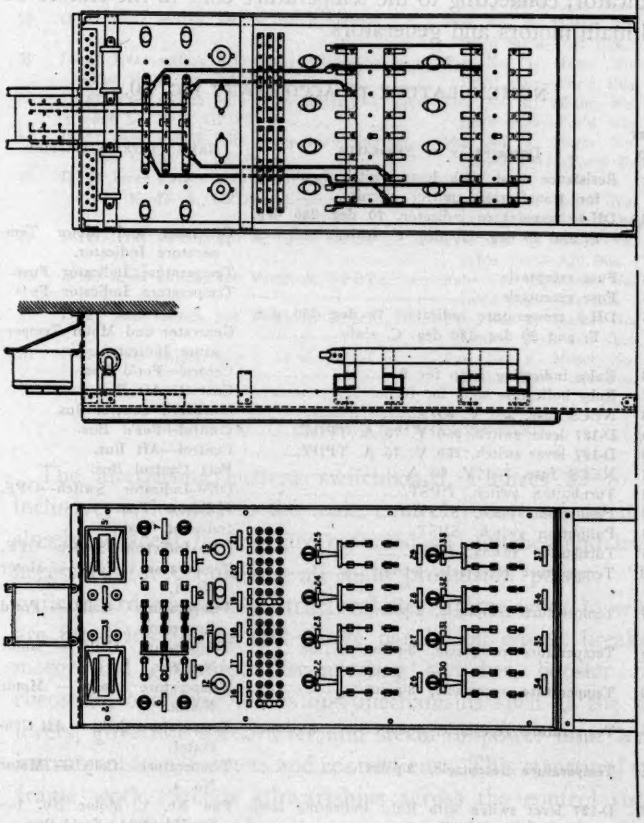


FIG. 30.—MOTOR VENTILATION CONTROL AND TEMPERATURE INDICATOR PANEL—U. S. S. "MARYLAND."

with indicating lamps: this switch being provided with current limiting resistance so that transfer from forward to after bus can be made without circuit interruption. Below this switch, on the panel are located the jacks for the temperature indicator, connecting to the temperature coils in the stators of all main motors and generators.

NOMENCLATURE TO ACCOMPANY FIG. 30.

Part No.	Description of Apparatus.	Name Plate Inscription.
1	Resistance used with lever switch 9 and 10 for transferring control circuit	
2	DH-6 temperature indicator, 70 deg.–250 deg. F. and 20 deg.–120 deg. C. scale	Generator and Motor Temperature Indicator.
3	Fuse receptacle	Temperature Indicator Fuse.
4	Fuse receptacle	Temperature Indicator Fuse.
5	DH-3 temperature indicator, 70 deg.–250 deg. F. and 20 deg.–120 deg. C. scale	Generator and Motor Temperature Indicator.
6	Ruby indicating lamp for 9	Control—For'd Bus.
7	Ruby indicating lamp for 10	Control—Aft Bus.
8	NECS fuse, 250 V. 60 A	Starboard Control Bus.
9	D-127 lever switch, 250 V. 75 A. TPDT	Control—For'd Bus.
10	D-127 lever switch, 250 V. 75 A. TPDT	Control—Aft Bus.
11	NECS fuse, 250 V. 60 A	Port Control Bus.
12	Turnbutton switch, DPST	ON—Indicator Switch—OFF.
13	Pullbutton switch, SPST	Indicator Test.
14	Pullbutton switch, SPST	Indicator Test.
15	Turnbutton switch, DPST	ON—Indicator Switch—OFF.
16	Temperature receptacle, 3-point	Temperature Coils — Motor No. 1.
17	Temperature receptacle, 3-point	Temperature Coils — For'd Generator.
18	Temperature receptacle, 3-point	Temperature Coils — Motor No. 2.
19	Temperature receptacle, 3-point	Temperature Coils — Motor No. 3.
20	Temperature receptacle, 3-point	Temperature Coils — Aft Generator.
21	Temperature receptacle, 3-point	Temperature Coils — Motor No. 4.
22	D-127 lever switch with Ruby indicating lamp 250 V. 75 A. DPDT	Fan No. 1, Motor No. 1—Outb'd Stb'd—For'd Bus.
23	D-127 lever switch with Ruby indicating lamp 250 V. 75 A. DPDT	Fan No. 2, Motor No. 1—Out'd Stb'd—For'd Bus.
24	D-127 lever switch with Ruby indicating lamp —250 V. 75 A. DPDT	Fan No. 1, Motor No. 2—Inb'd Stb'd—For'd Bus.

25	D-127 lever switch with Ruby indicating lamp—250 V. 75 A. DPDT	Fan No. 2, Motor No. 2—Inb'd Stb'd—For'd Bus.
26	D-127 lever switch, 250 V. 75 A. DPDT	Fan No. 1, Motor No. 1—Out'd Stb'd—Aft Bus.
27	D-127 lever switch, 250 V. 75 A. DPDT	Fan No. 2, Motor No. 1—Outb'd Stb'd—Aft Bus.
28	D-127 lever switch, 250 V. 75 A. DPDT	Fan No. 1—Motor No. 2—Inb'd Stb'd—Aft Bus.
29	D-127 lever switch, 250 V. 75 A. DPDT	Fan No. 2, Motor No. 2—Inb'd Stb'd—Aft Bus.
30	D-127 lever switch with Ruby indicating lamp—250 V. 75 A. DPDT	Fan No. 1, Motor No. 3—Inb'd Port—For'd Bus.
31	D-127 lever switch with Ruby indicating lamp—250 V. 75 A DPDT	Fan No. 2, Motor No. 3—Inb'd Port—For'd Bus.
32	D-127 lever switch with Ruby indicating lamp—250 V. 75 A. DPDT	Fan No. 1, Motor No. 4—Outb'd Port—For'd Bus.
33	D-127 lever switch with Ruby indicating lamp—250 V. 75 A. DPDT	Fan No. 2, Motor No. 4—Outb'd Port—For'd Bus.
34	D-127 lever switch, 250 V. 75 A. DPDT	Fan No. 1, motor No. 3—Inb'd Port—Aft Bus.
35	D-127 lever switch, 250 V. 75 A. DPDT	Fan No. 2, Motor No. 3—Inb'd Port—Aft Bus.
36	D-127 lever switch, 250 V. 75 A. DPDT	Fan No. 1, Motor No. 4—Outb'd Port—Aft Bus.
37	D-127 lever switch, 250 V. 75 A. DPDT	Fan No. 2, Motor No. 4—Outb'd Port—Aft Bus.

ALTERNATING CURRENT SWITCHBOARD.

The alternating current switchboard, Figures 32 to 38, inclusive, is located in the main control room and on it, or closely adjacent thereto, are mounted all switches and levers necessary for controlling all main propulsion power. This switchboard consists of structural steel frame work to which are fitted instrument and gauge panels, oil circuit breakers, motor and generator disconnecting switches, booster field rheostats, operating levers and mechanisms such as the field levers, governor speed lever and steam or power limit lever; instrument transformers and connections. This structural steel frame work extends athwartships across the control room: entrance to the back of the instrument panel and interior frame work, containing the oil circuit breakers, bus structure and disconnecting switches, being obtained through grille work doors at each side of the panel.

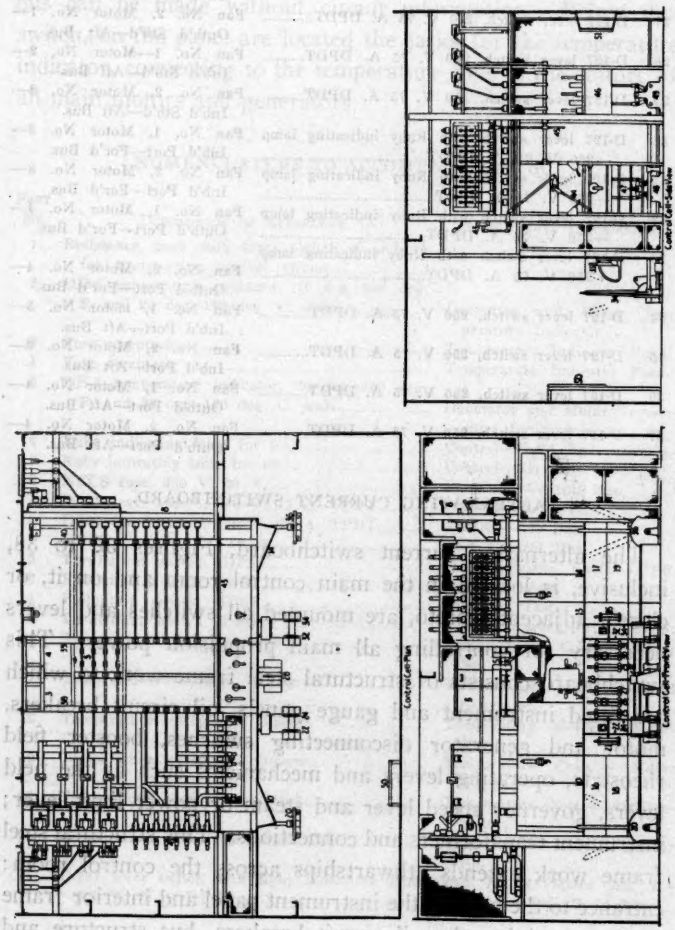


FIG. 32—A-C. CONTROL CELL—U. S. S. "MARYLAND."

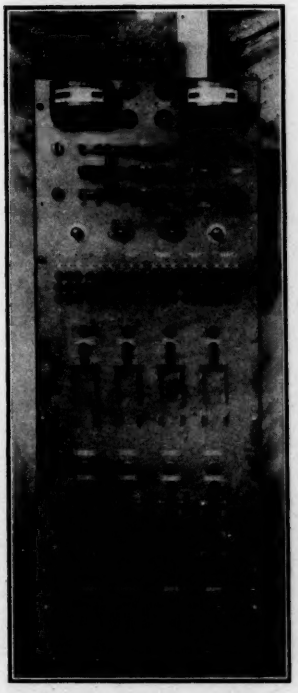


FIG. 31.—MOTOR VENTILATING CONTROL AND TEMPERATURE INDICATORS (FRONT VIEW) PROPULSION CONTROL EQUIPMENT U. S. S. "MARYLAND."

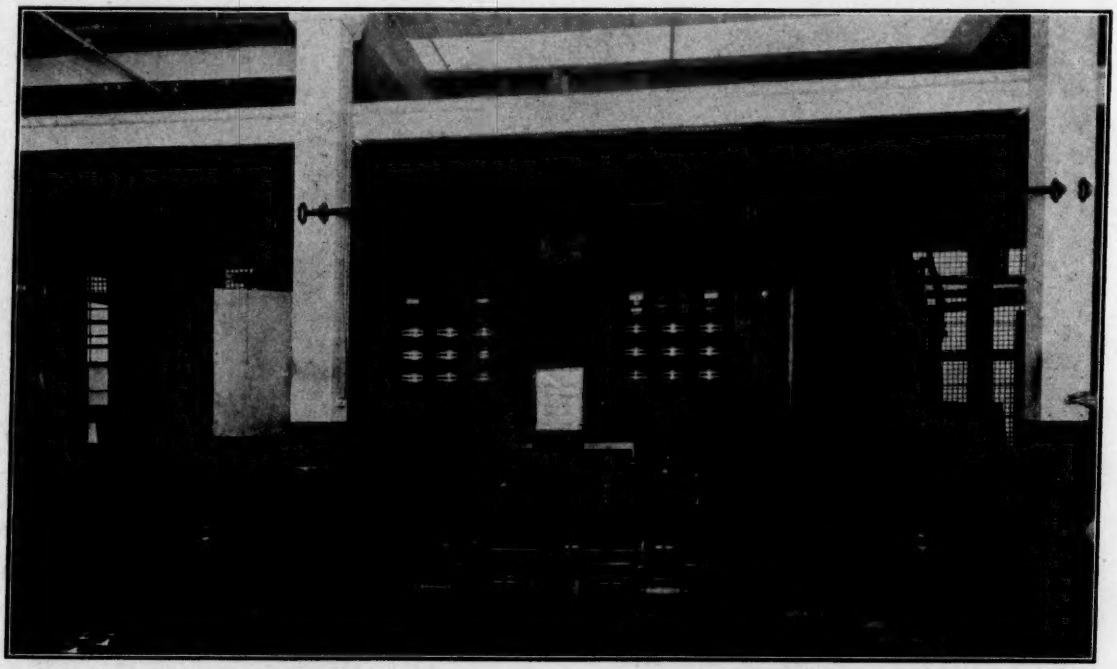


FIG. 33.—MAIN A. C. CONTROL BOARD. PROPULSION CONTROL EQUIPMENT U. S. S. "MARYLAND" AND U. S. S. "WEST VIRGINIA."

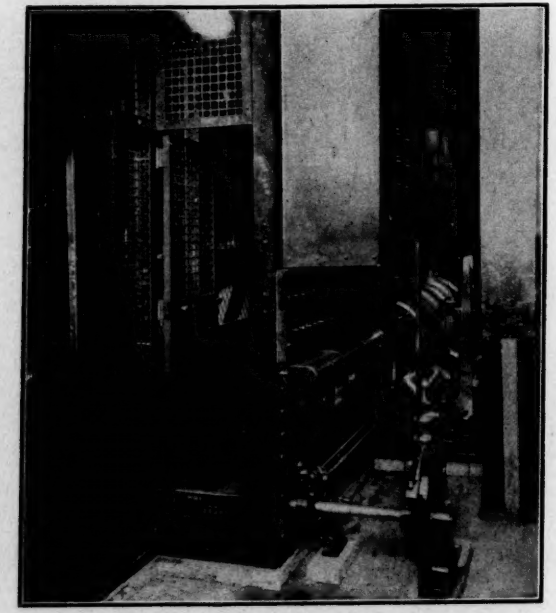


FIG. 34.—A. C. CONTROL CELL (SIDE VIEW) SHOWING AISLE BEHIND THE SWITCHBOARD. PROPULSION CONTROL EQUIPMENT U. S. S. "MARYLAND" AND U. S. S. "WEST VIRGINIA."

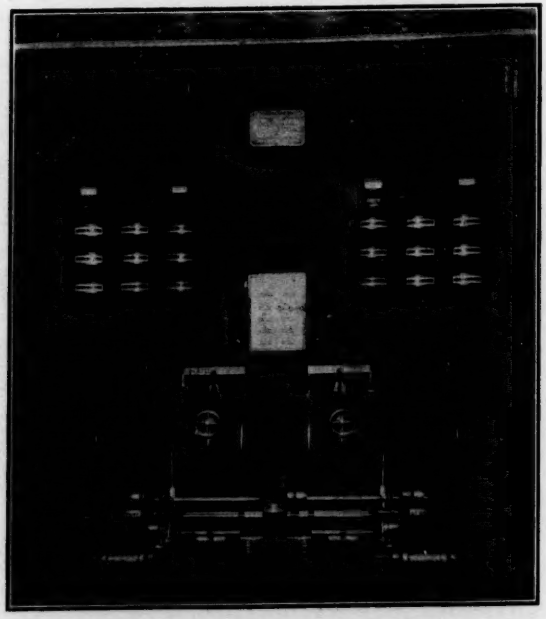


FIG. 40.—ELECTRICAL INSTRUMENTS AND OPERATING LEVERS FOR FIELD STEAM REVERSING. POLE CHANGING. FIELD SELECTIVE BUS TIE AND EMERGENCY TRIP. PROPULSION CONTROL EQUIPMENT U. S. S. "MARYLAND" AND U. S. S. "WEST VIRGINIA."

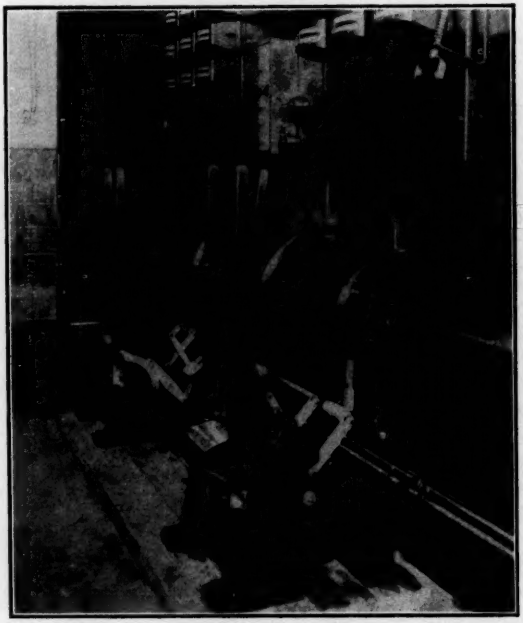


FIG. 41.—CONTROL LEVERS (SIDE VIEW). PROPULSION CONTROL EQUIPMENT U. S. S. "MARYLAND" AND U. S. S. "WEST VIRGINIA."

FIG. 42.—STEAM LIMIT. SPEED-FIELD AND FIELD SELECTIVE LEVERS AND INTERLOCK PROPULSION CONTROL EQUIPMENT. U. S. S. "MARYLAND" U. S. S. "CALIFORNIA" AND U. S. S. "WEST VIRGINIA."

On the front of the structural steel frame work are fitted ebony asbestos wood panels mounting instruments, meters, instrument transfer switches, indicating lamps, and revolution counters. At either side of these panels and set out at an angle from them are steel panels mounting the various steam gauges to each turbine. In the space immediately below the instrument and gauge panels are mounted the main test and terminal boards, balance and under current relays, a system of mechanical interlocks governing switching connections and their proper selection and the following levers; steam or power limit, governor speed, field control and field selective. It should be stated that each main generator field circuit breaker switch is located on the forward athwartship bulkhead of the control room and operated and closed by the first downward movement of the field lever mounted as indicated above.

CONTROL CELL U. S. S. "MARYLAND." FIG. 32.

Part No.	Description of Apparatus.
1	Generator Disconnecting Switch 4 PDT 5000 V. 2400 A.—Port.
2	Generator Current Transformers 3000 A.
3	Field Switch DPST 250 V. 400 A.—Port.
4	Field Switch DPST 250 V. 400 A.—Starboard.
5	Bus Tie Switch 4 PST 5000 V. 2400 A.
6	
7	
8	Motor Disconnecting Switch 8 PST. 5000 V. 600 A.—Outboard Starboard.
9	Emergency Trip for Main Turbine—Port.
10	Gauge Panel—Port.
11	Instrument Panel—Port.
12	Instrument Panel—Central.
13	Instrument Panel—Starboard.
14	Gauge Panel—Starboard.
15	Emergency Trip for Main Turbine—Starboard.
16	Testing and Terminal Board—Port.
17	Testing and Terminal Board—Starboard.
18	Balance Relay—Port.
19	Balance Relay—Starboard.
20	Operating Lever for Generator Disconnecting Switch.

21 Operating Lever for Field Selector Switch—Port.
22 Operating Lever for Pole Changing Oil Circuit Breaker—Port.
23 Under Current Relay—Port.
24 Operating Lever for Reversing Oil Circuit Breakers—Port.
25 Field Control Lever—Port.
26 Speed Control Lever—Port.
27 Steam Limit Lever—Port.
28 Operating Lever for Bus Tie Switch.
29 Steam Limit Lever—Starboard.
30 Speed Control Lever—Starboard.
31 Field Control Lever—Starboard.
32 Operating Lever for Reversing Oil Circuit Breaker—Starboard.
33 Under Current Relay—Starboard.
34 Operating Lever for Pole Changing Oil Circuit Breaker—Starboard.
35 Operating Lever for Field Selector Switch—Starboard.
36 Operating Lever for Generator Disconnecting Switch—Starboard.
37 Mechanical Interlocks.
38 Booster Field Rheostat—Port.
39 Booster Field Rheostat—Starboard.
40 Generator Disconnecting Switch 4 PDT 5000 V. 2400 A.—Starboard.
41 Motor Disconnecting Switch 8 PST 5000 V. 600 A.—Outboard Port.
42 Motor Disconnecting Switch 8 PST 5000 V. 600 A.—Inboard Port.
43 Motor Disconnecting Switch 8 PST 5000 V. 600 A.—Inboard Starboard.
44 Operating Lever for Motor Disconnecting Switch—Inboard Starboard.
45 Operating Lever for Motor Disconnecting Switch—Outboard Starboard.
46 Reversing Oil Circuit Breaker TPDT 5000 V. 3000 A.—Starboard.
47 Magnetic Lock for Oil Circuit Breaker.
48 Operating Mechanism for Pole Changing Oil Circuit Breaker.
49 Operating Mechanism for Reversing Oil Circuit Breaker.
50 Motor Ventilation Control Panel.
51 Terminal Board for Control Circuit.

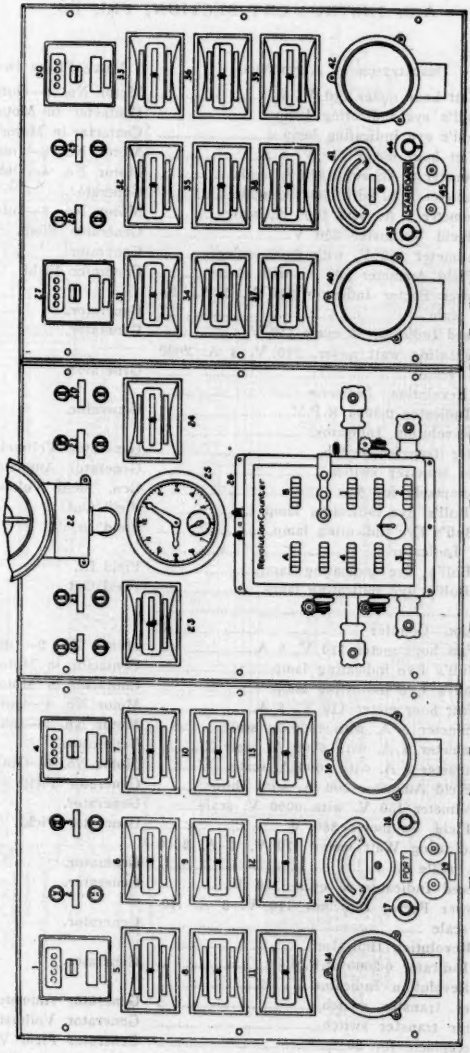
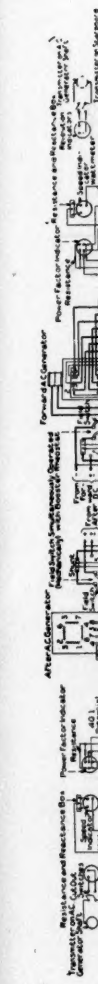


FIG. 35.—A-C INSTRUMENT PANELS AND GAUGE BOARD—U. S. S. "MARYLAND."

A.C. INSTRUMENT SECTION, FIG. 35.

Part No.	Description of Apparatus.	Name Plate Inscription.
1	IS-4 Watt hour meter 110 V. 5 A.	Motor No. 4—Outb'd Port.
2	Blue bull's eye indicating lamp	Contactor in Motor No. 4.
3	Blue bull's eye indicating lamp	Contactor in Motor No. 3.
4	IS-4 Watt hour meter 110 V. 5 A.	Motor No. 3—Inb'd Port.
5	H-2 Ammeter 5 A. with 1600 A. scale	Motor No. 4—Outb'd Port.
6	H-2 Ammeter 5 A. with 6000 A. scale	Generator.
7	H-2 Ammeter 5 A. with 1600 A. scale	Motor No. 3—Inb'd Port.
8	DH-3 Field Voltmeter 350 V.	Generator Field.
9	H-2 Voltmeter 150 V. with 6000 scale	Generator.
10	DH-3 Field Ammeter 500 A. with shunt	Generator Field.
11	H-2 Power Factor Indicator 110 V. 5 A. .40-1.00 scale	Generator.
12	H-4 Speed Indicator 25 cycle 110 V.	Generator.
13	H-2 Indicating watt meter, 110 V. 4 A. 2000 K.W. scale	Generator.
14	Shaft Revolution Indicator	
15	Speed Indicator 0-3000 R.P.M.	Generator.
16	Shaft Revolution Indicator	
17	Voltmeter transfer switch	Generator Voltmeter.
18	Ammeter transfer switch	Generator Ammeter.
19	Fuse receptacle for 8	Gen. Field Voltmeter.
20	Green Bull's Eye indicating lamp	Field Out.
21	Ruby Bull's Eye indicating lamp	Field In.
22	Rudder Indicator	
23	Ruby Bull's Eye indicating lamp	Field In.
24	Green Bull's Eye indicating lamp	Field Out.
25	Clock	
26	Revolution Counter	
27	IS-4 Watt hour meter 110 V. 5 A.	Motor No. 2—Inb'd Stb'd.
28	Blue Bull's Eye indicating lamp	Contactor in Motor No. 2.
29	Blue Bull's Eye indicating lamp	Contactor in Motor No. 1.
30	IS-4 Watt hour meter 110 V. 5 A.	Motor No. 1—Outb'd Stb'd.
31	H-2 Ammeter 5 A. with 1600 A. scale	Motor No. 2—Inb'd Stb'd.
32	H-2 Ammeter 5 A. with 6000 A. scale	Generator.
33	H-2 Ammeter 5 A. with 1600 A. scale	Motor No. 1—Outb'd. Stb'd.
34	DH-3 Field Ammeter 500 A. with shunt	Generator Field.
35	H-2 Voltmeter 150 V. with 6000 V. scale	Generator.
36	DH-3 Field Voltmeter 350 V.	Generator Field.
37	H-2 Indicating Watt meter 110 V. 4 A. 2000 K.W. scale	Generator.
38	H-4 Speed Indicator 25 cycle 110 V.	Generator.
39	H-2 Power Factor Indicator 110 V. 5 A. .40-1.00 scale	Generator.
40	Shaft Revolution Indicator	
41	Speed Indicator 0-3000 R.P.M.	Generator.
42	Shaft Revolution Indicator	
43	Ammeter transfer switch	Generator Ammeter.
44	Voltmeter transfer switch	Generator Voltmeter.
45	Fuse receptacle for 36	Generator Field Voltmeter.



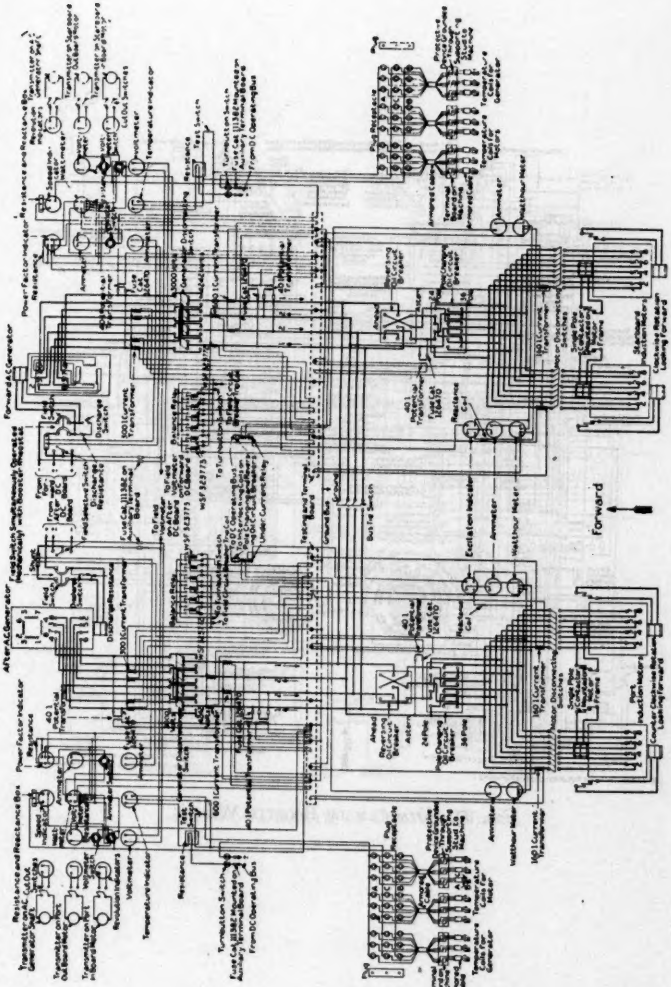


FIG. 36.—DIAGRAM OF A-C. CONTROL WIRING.

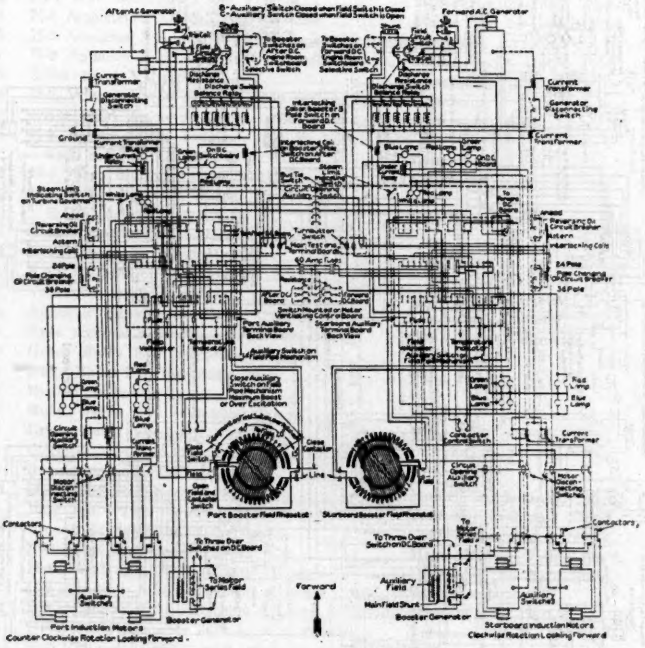


FIG. 37.—DIAGRAM OF BOOSTER WIRING.

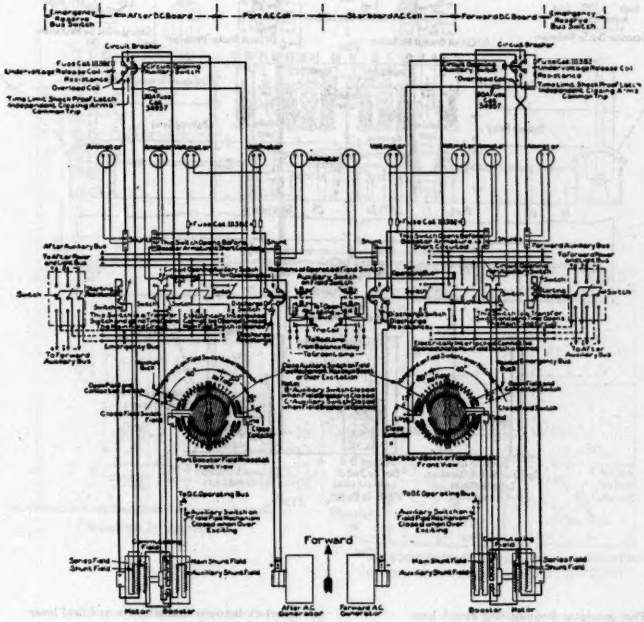
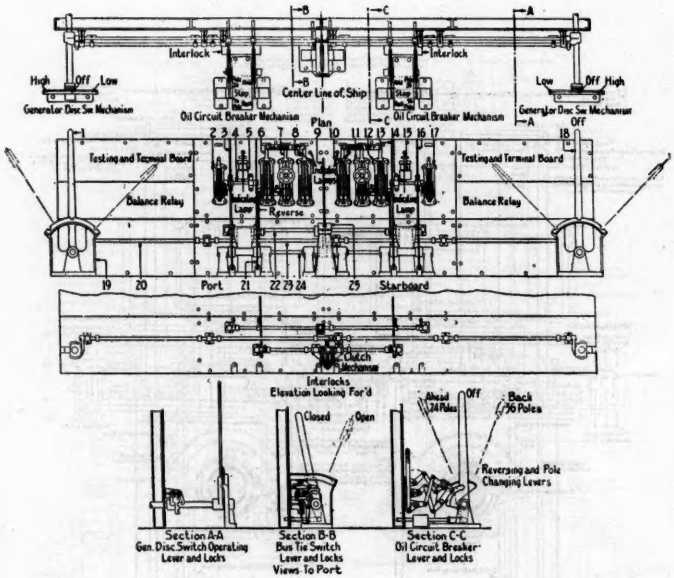
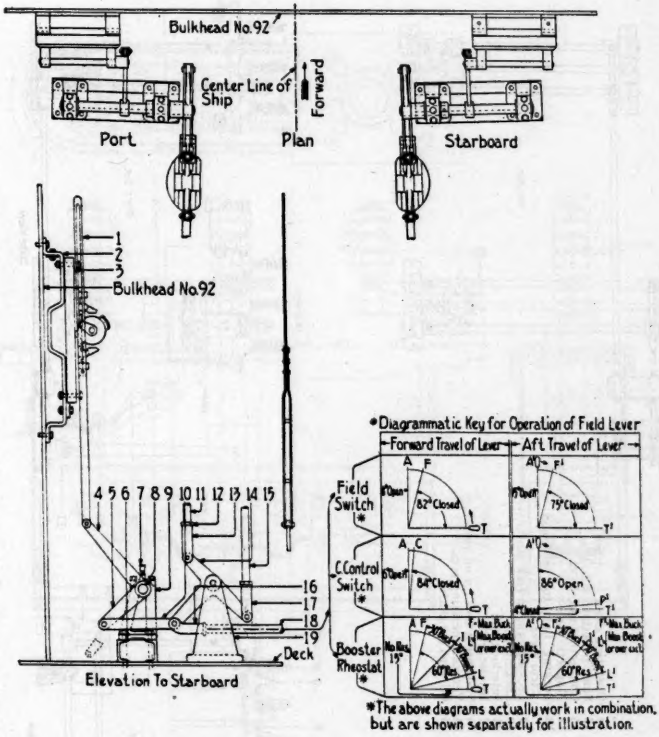


FIG. 38.—DIAGRAM OF CONTROL WIRING.



1 Port generator disconnecting switch lever
2 Port field selective lever
3 Pole changing switch lever
4 Interlock between field selective and field lever
5 Under current relay
6 Field control lever
7 Speed control lever
8 Steam limiting lever
9 Bus tie switch lever
10 Steam limit lever
11 Speed control lever
12 Reversing lever
13 Field control lever
14 Interlock between field selective and field lever
15 Under current relay
16 Pole changing switch lever
17 Field selective lever
18 Starboard generator disconnecting switch lever
19 Pedestal for generator disconnecting switch shaft
20 Interlocking shaft
21 Reversing lever
22 Interlocking shaft
23 Interlocking shaft
24 Interlocking shaft
25 Bus tie switch index plate

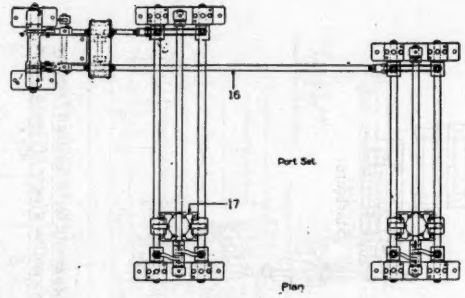
FIG. 39.—ARRANGEMENT OF CONTROL LEVERS.



1 Slotted link
2 Bracket for contactor switch
3 Contactor control switch
4 Crank
5 Crank
6 Foundation
7 Grease cup
8 Shaft
9 Shaft bearing
10 Link
11 Connecting rod to field switch
12 Lock nut
13 Coupling
14 T-shaped crank
15 Connection rod to booster field rheostat
16 Coupling
17 Coupling
18 Connecting rod for field lever
19 Hanger

FIG. 46.—ASSEMBLY OF OPERATING MECHANISM FOR CONTACTOR CONTROL SWITCH—U. S. S. "MARYLAND."

1 Top frame
2 Reversing lever
3 Pole changing lever
4 Locking crank
5 Locating springs
6 Pivot shaft pedestal
7 Couplings
8 Adjusting screw
9 Hanger
10 Stud
11 Link
12 Bell crank
13 Oil vessel
14 Operating shaft
15 Shaft pedestal
16 Connecting rod
17 Magnetic lock
18 Locking link to field control lever
19 Locking cam on lock shaft
20 Locking dog
21 Interlock box
22 Insulator
23 Suspension bolt for oil vessel
24 Vertical operating rods
25 Oil cups for shafts



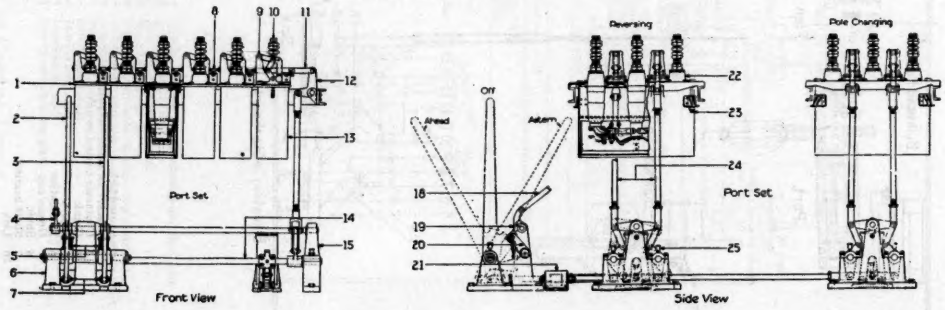
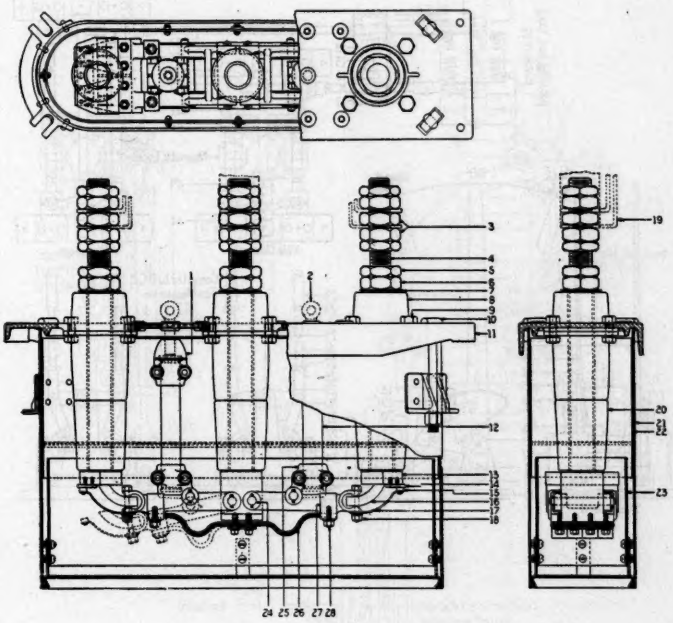
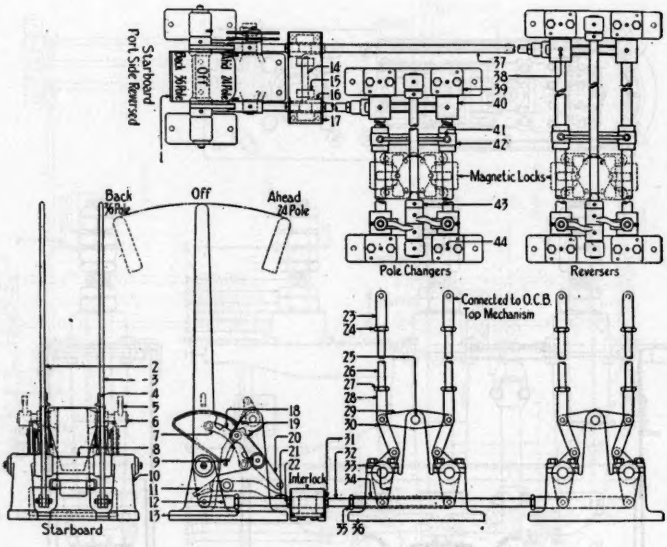


FIG. 47.—ARRANGEMENT OF OPERATING MECHANISM FOR OIL CIRCUIT BREAKERS.



1 Clevis coupling
2 Screw eye
3 Contact nut
4 Stud
5 Lock nut
6 Clamp nut
7 Washer
8 Insulator
9 Mounting bolt for insulator
10 Tank suspension bolt
11 Top frame
12 Wood insulating rod
13 Contact block
14 Burning plate
15 Burning tip
16 Main brush contact
17 Secondary finger
18 Clamp bolt for main brush
19 Connection bars
20 Insulating bushing
21 Oil vessel
22 Insulating lining—upper
23 Insulating lining—lower
24 Hinge pin for contact arm
25 Coupling
26 Laminated connection
27 Contact arm
28 Attaching screws for laminated connection

FIG. 48.—ASSEMBLY OF OIL CIRCUIT BREAKER.



1	Index plate support
2	Reversing lever
3	Pole changing lever
4	Index finger
5	Cranks for interlocking
6	Shaft
7	Notched sector
8	Index plate
9	Spring for locking dog
10	Shaft bearing
11	Kick off lever
12	Coupling
13	Lock nut
14	Bevelled bolt
15	Guides
16	Bevelled blocks
17	Interlock box
18	Notched quadsant on lever
19	Locking cam
20	Interlock lever
21	Connecting rod
22	Lock nut
23	Coupling
24	Lock nut
25	Fin shaft
26	Connecting rod
27	Lock nut
28	Coupling
29	Aligning cranks
30	Toggle link
31	Lock nut
32	Connecting rod
33	Operating crank
34	Connecting link
35	Lock nut
36	Shaft pedestal
37	Connecting rod to pole changing switch
38	Set screw
39	Bearing cap
40	Operating crank
41	Grease cup
42	Shaft spacer
43	Collar
44	Grease cup

FIG. 49.—ARRANGEMENT OF OPERATING MECHANISM FOR POLE CHANGING AND REVERSING SWITCH.

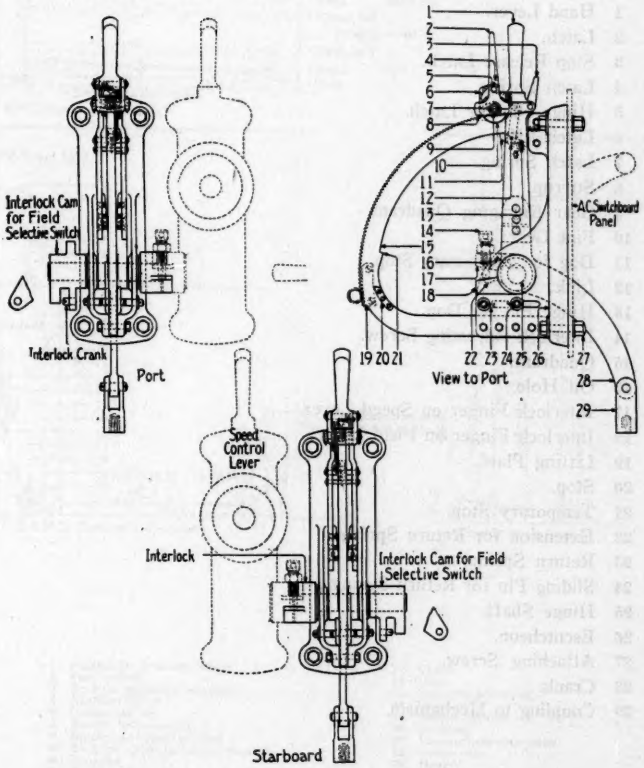
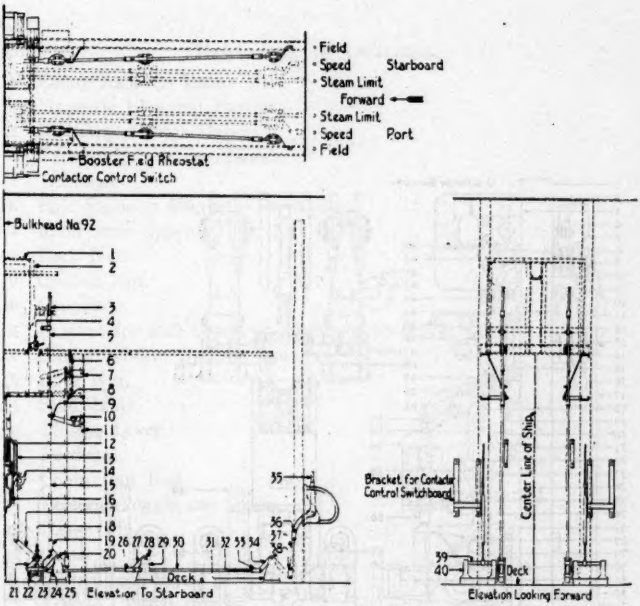


FIG. 43.—ASSEMBLY OF FIELD CONTROL LEVER

FIELD CONTROL LEVER, FIG. 43.

Part No.	Description of Apparatus.
1	Hand Lever.
2	Latch.
3	Stop Release Latch.
4	Latch Spring.
5	Hinge Pin for Latch.
6	Latch Lift.
7	Latch Spring.
8	Stirrup.
9	Gear Engaging Quadrant.
10	Fine Gear.
11	Dog for Temporary Stop.
12	Link.
13	Hinge Pin for Dog.
14	Interlock Adjusting Screw.
15	Quadrant.
16	Oil Hole.
17	Interlock Finger on Speed Lever.
18	Interlock Finger on Field Lever.
19	Lifting Plate.
20	Stop.
21	Temporary Stop.
22	Extension for Return Spring.
23	Return Spring.
24	Sliding Pin for Return Spring.
25	Hinge Shaft.
26	Escutcheon.
27	Attaching Screw.
28	Crank.
29	Coupling to Mechanism.



1 Bracket for field switch base
2 Field switch
3 Rack for operating field switch
4 Auxiliary switch
5 Connecting rod
6 Booster field rheostat
7 Bracket and guide for rack
8 Rack for operating booster rheostat
9 Coupling
10 Guide arm
11 Hinge bracket for guide arm
12 Auxiliary return spring
13 Slotted link
14 Contactor control switch
15 Connecting rod
16 Bracket for contactor control switch
17 Crank
18 Grease cup
19 Coupling
20 Grease cup
21 Crank
22 Link
23 T-shaped bell crank
24 Coupling
25 Connecting rod
26 Coupling
27 Intermediate suspension
28 Grease cup
29 Hanger
30 Connecting rod
31 Hanger
32 Lock nut
33 Coupling
34 Bell crank
35 Field control lever
36 Coupling
37 Connecting rod
38 Lock nut
39 Counter shaft
40 Shaft bearing

FIG. 44.—ARRANGEMENT OF FIELD CONTROL MECHANISM—U. S. S. "MARYLAND."

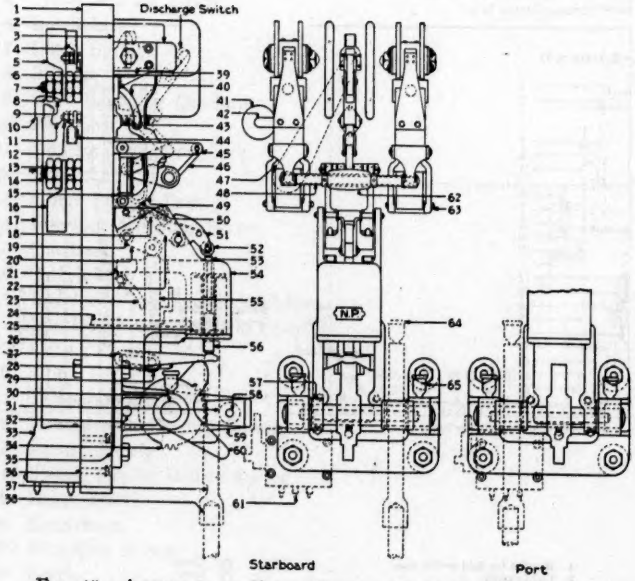


FIG. 45.—ASSEMBLY OF FIELD SWITCH—U. S. S. "MARYLAND."

FIELD SWITCH, FIG. 45.

Part No.	Description of Apparatus.
1	Ebony Asbestos Base.
2	Magnetic Blow-out Barrier.
3	Discharge Switch Barrier.
4	Terminal-Discharge Switch.
5	Terminal.
6	Pole Piece on Magnetic Blow-out.
7	Main Stud Upper.
8	Lock Nut.
9	Contact Nut.
10	Coupling.
11	Bracket for Bell Crank on Auxiliary Switch.
12	Stud Discharge Switch.
13	Lock Nut.
14	Stud, Main.
15	Closing Lever.
16	Terminal.
17	Connecting Rod.
18	Locking Toggle and Spring.
19	Hinge Pin.
20	Crank.
21	Spring for Finger.
22	Toggle Closing Finger.
23	Finger.
24	Insulating Tube for Trip Coil Leads.
25	Spool.
26	Cover Plate.
27	Guide for Plunger.
28	Trip Lever.
29	Roller.
30	Shaft.
31	Roller for Rack.
32	Alignment Pin.
33	Auxiliary Switch 3 P.S.T.
34	Gear Keyed to Shaft.
35	Attaching Bolt.
36	
37	Rack.
38	Connecting Rod to Field Mechanism.
39	Main Contact Block.
40	Main Brush.

41 Auxiliary Switch Escutcheon.
42 Operating Finger for Auxiliary Switch.
43 Auxiliary Switch Plunger.
44 Terminal on Discharge Switch Stud.
45 Link.
46 Bell Crank.
47 Connection Bar.
48 Cross Bar.
49 Main Brush Lever.
50 Laminated Connection.
51 Trip Lever.
52 Roller.
53 Tripping Plunger.
54 Magnet Frame.
55 Closing Plunger.
56 Tripping Plunger Extension.
57 Collar on Shaft.
58 Pin and Cotters for Roller.
59 Shaft Bearing and Rack Guide.
60 Closing and Tripping Crank.
61 Connections for Auxiliary Switch.
62 Operating Crank.
63 Hinge Pin for Main Brush Lever.
64 Grease Cup for Rack.
65 Grease Cup for Shaft.

ENGINE ROOM OPERATION.

In the engine room the following is the sequence of operation at the D.C. engine room switchboard:

(1) Start the D.C. generator selected to supply engine room auxiliaries and excitation for the A.C. generator. It should be emphasized that, for obvious reasons, the light and power of the ship is never paralleled with main propulsion D.C. power.

(2) Adjust field of D.C. generator until normal voltage of 240 is obtained.

(3) Close triple pole single throw lever switch connecting this generator to auxiliary bus.

(4) Close the auxiliary motor feeder switches into upper throw thus connecting them to auxiliary bus. Then start up the various main engine auxiliaries at their respective starting panels.

(5) Start the main turbine—This should be allowed to run one and a half hours for proper warming up to secure uniform temperature conditions throughout, at a speed of about 600 R.P.M.

(6) Close feeder switch to motor ventilation control panel in upper throw.

(7) Close triple pole circuit breaker in the tie line to other engine room switchboard.

(8) Close triple pole double throw emergency bus transfer switch. If both engine rooms boards are alive this switch should be closed on the tie line from other engine room board. If only one board is alive throw the switch to power and light bus connections thus energizing the auxiliary reserve bus.

(9) Close the booster motor circuit breaker.

(10) Close the pump type starting switch to the booster, starting same.

(11) Close the double pole double throw broken back switch in the upper throw.

(12) Close the triple pole, double throw field switch on upper throw and see that magnetic lock holding it is operative. From this point the propulsion equipment is controlled from the alternating current switchboard in the control room.

(13) In case of trouble with the booster and only after the A.C. generator field switch is open in the control room, throw the double pole double throw broken back switch to lower throw and trip the booster motor circuit breaker. All auxiliary feeders and feeder to the ventilation panel in the control room should now be transferred to auxiliary reserve bus by throwing to the lower throw, restarting auxiliary motors at their respective starting panels if necessary, although this can be done without the motors stopping. Under this condition the A.C.

generator field is no longer affected by the booster field rheostat, but may be varied by manipulation of the field rheostat hand wheel of the D.C. generator connected to the auxiliary bus.

CONTROL ROOM OPERATION.

Before giving the sequence of operation in the control room it is clarifying to compare the operating levers of electric drive to the main features of common automobile drive which are most familiar. The main field lever of the control room is the clutch; the governor speed lever is the automobile accelerator; the pole changing lever is the low and high speed selective transmission lever; and the reversing lever corresponds to the automobile transmission lever thrown to its backing position. Viewed in the above light, electric drive operation from the control room is most easily understood and essentially simple.

The usual running speeds are listed in the following table and the manipulation of apparatus follows in detail. See Figures 39 to 49, inclusive.

SPEED CONDITIONS.

Condition.	Speed in knots	Number of generators.	Connection of generators.	No. of motors used.	Pole conn. of motors.	Position of bus tie switch.
1	0–15	1	Low voltage	4	36	Closed
2	15–17	1	Low voltage	4	24	Closed
3	17–full speed	2	High voltage	4	24	Open
4	Backing about 10.	1	Low voltage	4	36	Closed
5	Backing	2	High voltage	4	36	Open

CONDITION 1.

(*a*) All switches open except motor disconnects.

(*b*) Close bus tie or paralleling switch.

(*c*) Close selected generator disconnecting switch on its low voltage throw.

(*d*) Throw both pole changing levers to "36 pole" position.

(*e*) Throw reversing levers to "Ahead" position.

(*f*) Pull down field lever to its first stop thereby closing main generator field switch and producing over excitation for pulling motors into step. Watch motor revolution indicators carefully and when these reach a steady running indication of about 45 R.P.M. release stop on field lever and move lever to its extreme position downward, thus closing contactor control switch and putting motors on their running high-power factor copper outer windings. That the field is closed and the motor contactors throw in is shown by indicating lights on the top of the board.

(*g*) Adjust field excitation to a value sufficient to keep the motors in step, which value will be clearly indicated by the "excitation indicator."

(*h*) Adjust speed lever for proper turbo generator speed corresponding to the desired R.P.M. of the propellers.

(*i*) Along with operation (*h*) adjust steam or power limit lever to illuminate the white indicating light just above this lever. However, this adjustment of steam limit should only be done, as indicated hereinbefore, when steady running conditions are reached. Where maneuvering is probable, such as in rivers and harbors or during fleet exercises, wide speed changes must be available without necessity of adjusting steam limit.

CONDITION 2.

The operation in condition 2 is exactly the same as for condition 1 up to and through (*g*).

(*h*) As soon as a speed of approximately 8 knots is reached and ship has good way upon her,

(1) Throw off field switch.

(2) Bring governor speed lever to vertical position or lowest running speed.

(3) When under-current relay releases magnetic locks, as shown by its indicating light going off, release the mechanical interlock between reversing and pole changing levers by means of foot pedal and

(4) Throw pole changing levers to "24 pole" connection. Thereafter the procedure is repeated as in (*f*), (*g*), (*h*) and (*i*) of condition (1).

CONDITION 3.

Condition 3 procedure is exactly the same as condition 2 procedure except that the bus tie or paralleling switch in (*b*) is opened and both generator disconnects of (*c*) are thrown to the high voltage connection.

CONDITION 4.

Condition 4 procedure is exactly the same as condition 1 procedure except that the reversing levers of (*e*) are thrown to the "Back" connection.

CONDITION 5.

Condition 5 is the same as condition 4 except that both generator disconnects are thrown to the "high" voltage connection, it being emphasized that only 36 pole connection must be used in backing.

OPERATION IN CASE OF FAILURE OF ONE EXCITER.

In all cases it is assumed that the main generator field switch is thrown off to its vertical position at the main control board; then the following operations occur at the engine room direct current switchboards.

CONDITION 1.

The assumption is the remaining two exciters controlled by same switchboard are in operating condition.

(*a*) Cut all field rheostat resistance into field of disabled exciter.

(*b*) Open triple pole single throw lever switch connecting this exciter to auxiliary bus.

(*c*) Close the open triple pole single throw lever switch of the exciter feeding the light and power bus at the time, it being emphasized that such a generator must be on light and power load at all times.

(*d*) Restart auxiliary motors at their respective starting panels.

(*e*) Start third exciter and parallel it with running exciter on power and light bus.

(*f*) Disconnect exciter which is feeding the auxiliary bus from the power and light bus.

CONDITION 2.

The assumption is that only one of the remaining exciters is in operation condition.

(*a*) Throw feeder switches of main engine auxiliaries to the auxiliary reserve bus.

(*b*) Restart auxiliary motors at their starting panels.

RELAYS.

Under current relays are fitted to operate by means of instrument transformers placed in the main motor circuits so that, after the main A.C. generator field switch is opened, neither the pole changing nor reversing levers may be opened until the motor currents have dropped to a safe opening value. This relay is fitted, of course, to protect the pole changing and reversing oil circuit breakers. A short circuiting pull-button type switch is provided on the relay cover side so that, in

an emergency requiring quick action, the relay may be shorted and hence the pole changing and reversing levers moved at will.

Balance relays are fitted in each generator circuit for protection against short circuits and grounds in the windings or cables attached thereto. Each of the two relays consists of six units, electrically separate but mechanically interconnected through link mechanism. Each unit consists of a vertically moving plunger. When the latter is down it closes a set of contacts which in turn open up the main field switches through a field trip coil. Connections to these relays are shown in Figure 36.

LOCKS AND INTERLOCKS.

The system of locks and interlocks are shown in Figure 50.

(1) The governor speed lever is provided with an adjustable stop preventing its operation in advance of the field lever.

(2) Similarly a field lever stop prevents the lever being thrown to its low position, until after the stop has been lifted.

(3) The field selective switch is mechanically interlocked with lever for operating field circuit breaker so that former is inoperative unless field circuit breaker is open.

(4) The booster field rheostat and field circuit breaker are interlocked inasmuch as their mechanism is operated simultaneously by same lever.

(5) The main field lever switch on each engine room D.C. board is electrically interlocked with field circuit breaker so that it may not be opened until circuit breaker is open. However the lever switch can be closed independently of circuit breaker closure.

(6) The motor contactors control switch for operating motor running windings is operated by field circuit breaker mechanism and hence, in effect, interlocked with the latter.

(7) The motor disconnecting switches are interlocked with their respective reversing oil circuit breakers so that the former may not be operated unless the latter are open.

(8) The bus tie or paralleling switch and generator disconnects are so interlocked that:

(*a*) With bus tie switch open generator disconnects can be thrown into high voltage connection only.

(*b*) With the bus tie switch closed either of generator disconnects can be thrown into low voltage position only.

(9) The bus tie switch cannot be opened when both generator disconnects are closed in the low voltage position.

(10) The bus tie and generator disconnects cannot be operated unless reversing lever is on off position.

(11) The pole changing and reversing levers are inoperative until motor currents have fallen to predetermined value controlled by under current relay.

(12) Pole changing and reversing levers are interlocked with field lever so that they cannot be operated unless field lever is "Off."

(13) Pole changing and reversing levers are interlocked with field lever so that, when either field lever is used alone, the pole changing and reversing levers on opposite side are also locked against operation.

(14) The pole changing and reversing levers are interlocked with the bus tie lever so that the former cannot be operated unless the latter is in either extreme position.

(15) The pole changing and reversing levers are interlocked so that the reversing lever cannot be thrown into "Ahead" position unless the pole changing lever is in the "36 pole" or "Off" position. However, this interlock may be relieved at will by a foot pedal. This device prevents the accidental establishment of the 24 pole connection for starting in as much as this only provides for comparatively low torque.

(16) Pole changing and reversing levers are so interlocked that the reversing lever cannot be thrown "Astern" unless pole changing lever is on "36 poles." Also the pole changing lever cannot be thrown into the 24 pole position when reversing lever is "Astern."

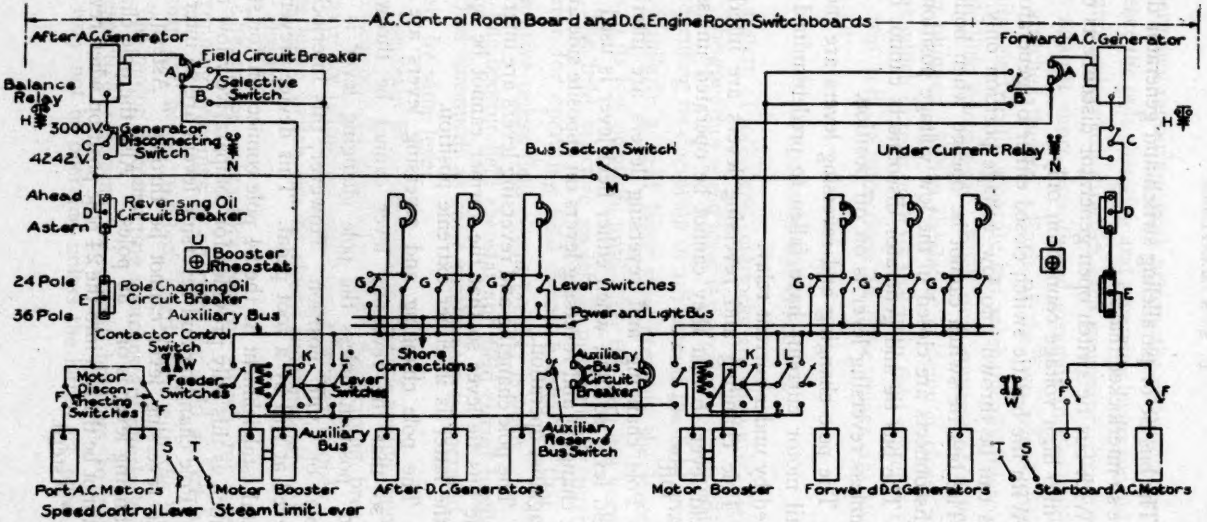


FIG. 50.—LOCKS AND INTERLOCKS.

SHAFTING AND BEARINGS.

There are four lines of shafting each driven by the main propelling motors, each line of shafting being composed of the following number of separate sections all rigidly coupled together:

	Inboard shafts.	Outboard shafts.
Thrust shaft.....	1	1
Line shaft.....	2	2
Stern tube shaft.....	1	1
Propeller shaft.....	1	1

Thrust Shafts:—Thrust shafts are fitted with thrust bearings of the Kingsbury type, having a steady bearing near the after coupling and at each end of the thrust bearing. Coupling discs and thrust collar are forged integral with the shaft.

Line Shafts:—These are hollow forged with disc couplings at each end forged integral with the shaft. The after section of the outboard line shafting and both sections of the inboard line shafting are carried in two steady bearings, one near each end of the shaft. The forward section of the outboard line shafting is carried by a single steady bearing just forward of its center.

Stern Tube Shafts:—These are hollow forged and secured to the line shafts through special disc couplings so that the shaft may be removed through the stern tube bearings. Composition casings are shrunk on the shaft where they pass through stern tube and stern tube bearings. Special sleeve couplings are fitted to secure the stern tube shafting. Lignum Vitae bearings are provided. A standard stuffing box gland is provided where shaft passes through the forward bulkhead at frame 115 for the outboard shafts and frame 124 for the inboard shafts. Fairways are fitted where the shaft leaves the stern tube bearings.

Propeller Shafts:—These are hollow forged, taper turned at after end to suit propeller hub and have composition casing shrunk on where they pass through the strut bearing, each shaft being carried by a single Lignum Vitae line strut bearing. Fair waters are fitted at each end of this bearing.

Inboard Couplings:—The stern tube shaft is upset at its inner end and has fitted over this upset portion a sleeve secured by four keys. At rear of the sleeve a collar, made in halves, is fitted against the shoulder formed by the upsetting, these collars being secured to the sleeve and line shaft disc by eight through bolts.

Outboard Couplings:—These couplings are of the solid sleeve type, taper bored for the reception of the shafts, secured to the shafts by two feather keys (axial) and one cross key (radial).

Where line and thrust shafts pass through water tight bulkheads a special type sleeve stuffing box is provided. A sleeve with standard type stuffing box gland fitted in its end and supported from the shaft foundation is fitted in the bulkhead. By this arrangement bulkhead deflection may occur without binding the shaft.

SHAFT DATA.

Thrust shafts, length, inboard, feet and inches	22–2
length, outboard, feet and inches	22–2
diameter, inches	15¼
diameter at bearings, inches	15½
diameter axial hole, inches	7⅝
thrust collars, number	1
thrust collar thickness, inches	6½
thrust collar outside diameter, inches	30⅜
thrust collar inside diameter, inches	15.74
Line shafts, forward, inboard, length, feet and inches	28–8¾
after, inboard, length, feet and inches	28–8¾
forward, outboard, length, feet and inches	19–8
after, outboard, length, feet and inches	19–8
diameter, inches	15¼
diameter at bearings, inches	15½
diameter axial hole, inches	7⅝

Stern tube shafts, length, feet and inches	41–9
diameter, inches	16⅞
diameter at couplings, inches	18⅜
diameter at bearings, inches	18⅝
diameter axial hole, inches	8⅜
diameter axial hole at outboard coupling sleeve, inches	4
Sleeve, thickness at bearings, inch	⅞
Taper in outboard couplings, inches per foot	1
Propeller shaft, outboard, length, feet and inches	34–0
Propeller shafts, inboard, length, feet and inches	31–0
diameter, inches	16⅞
diameter, axial hole, inches	8⅜
diameter axial hole at propeller hub and outboard coupling, inches	4
taper in outboard coupling, inches per foot	1
taper in propeller hub, inches per foot	1
Coupling discs, diameter, inches	22½
number, bolts	8
Inboard couplings, diameter of sleeve, outside, inches	30
diameter of sleeve, outside, inches	18⅜
length of sleeve, inches	13
thickness of collar, inches	3½
Coupling bolts, number, each coupling	8
diameter (taper) at face of coupling, inches	2¾
Outboard coupling, length of sleeve, inches	63¼
diameter, outside, inches	22½

REFRIGERATING SYSTEM.

The ice machine room is situated between frames 60 and 68 on the starboard side of the platform deck and has installed therein three ice machines of the CO2 type with a capacity of 3 tons each, the compressors of which are driven by reciprocating steam engines; all of the equipment is supplied by Kroeschell Brothers Ice Machine Co. Two Worthington-Blake-Knowles simplex brine pumps, size 6 inches × 5¾ inches × 6 inches are installed for brine circulation. Two exactly similar pumps are installed for sea water circulation. There are also fitted three condensers for the CO2 gas; one brine storage tank; one ice making tank of three compart-

ments; two brine cooled scuttle-butts, one each located on main and second deck; cold storage room for meats, eggs and cheese, cream and butter, and cooling room for officers mess; all located on third deck; and a 40-quart ice cream freezer and hardening box located on main deck.

EVAPORATOR PLANT.

As originally installed this plant was of the high pressure type. It consisted of three first effect evaporators, three second effect evaporators, three coil drain feed water heaters, two vapor feed water heaters, four distillers, two evaporator feed pumps, two fresh water pumps, two distiller circulating pumps, one brine pump, one distiller air pump and a Cochrane precision meter to measure the output. All this equipment was located in the evaporator space, betwen frames 60 and 68 on the first and second platforms. The above is now being replaced with a low pressure installation built at the Navy Yard, New York; it is designed to operate on exhaust steam and should effect a very considerable improvement in economy.

GENERAL WORK SHOP.

The following machine tool equipment, all electric driven, is installed in the General Work Shop located on the third deck over the forward engine room:

1—48-inch Gap Lathe.
Type—Rahn Larno Co., Series G, Lot 12.
Size—No. 24-48 inches × 12 inches.
Motor—G. E. Co., 7.5 H.P.; Volts 230; 450/1350 R.P.M.; Shunt.

1—Shaper.
Type—American Tool Works Co.
Size—24 inches.
Motor—Reliance Electric Co., 3 H.P.; Volts 230; 1150 R.P.M.; Shunt.

2—14-inch Lathes.
Type—Tool Makers; American Tool Works Co.
Size—No. 1.
Motor—Reliance Electric Co., 3 H.P.; Volts 230; 1150 R.P.M.; Shunt.
1—Hand Drill Press.
Type—Willey Electric Drill.
Size—16.
Motor—J. Clark, Jr., Electric Co., ½ H.P.; Volts 230.
1—Radial Drill Press.
Type—American Drill Press Tool Co.
Motor—Reliance Co., 3 H. P.; Volts 230; 1700/1800 R.P.M.; Shunt.
1—Precision Lathe.
Type—Rivett Lathe and Grinder Co.
Motor—Reliance Electric Co., 2¾ H.P.; Volts 230; 1700/1800 R.P.M.; Shunt.
1—Electric Grinder (stationary).
Type—Willey Electric Grinder.
Motor—J. Clark, Jr.; 1700 R.P.M.; Volts 230.
1—Milling Machine.
Type—Universal; The Kempsmith Mfg. Co.
Motor—Reliance Electric Co., 3 H.P.; 230 Volts; 1150 R.P.M.; Shunt.
1—Electric Hack Saw.
Type—No. 17730.
Motor—J. Clark, Jr.; 230 Volts; 2 Amps.
2—Electric Grinders (portable type).
Type—D3.
Motor—J. Clark Co.; 115 Volts; 3600 R.P.M.

COPPERSMITH SHOP, BLACKSMITH SHOP AND FOUNDRY.

The coppersmith shop, blacksmith shop and foundry are located in one compartment, between frames 72 and 79, on the port side of the upper deck and contain the following equipment:

1—Bending Slab, Navy Type, size 6 feet × 3 feet.
1—Pipe Threading Machine, Electric Driven, Mfg. by Curles & Curles Co.
1—Cupola—Navy Standard; 30 inches high; 44 inches diameter.
1—Hand Punch; P. S. & W. Co.
1—Forge, Coppersmith, Hauck Burner; ¼ inch opening.
1—Forge, Blacksmith, Hauck Burner; ¼ inch opening.
1—Set Rolls, Niagara Mfg. Co., 30 inches rollers.
1—Beading Machine, Hand, Niagara Mfg. Co., 11 inches rollers.
1—Folder Brake; Peck, Stow & Wilcox Co., 4 inches rollers.
1—Burring Machine, Hand; P. S. W. Co.
1—Torch, Portable, 15-gallon Tank; Hauck Patent Burner.

The thrust bearings, propellers, communication system, general piping arrangement, boilers, boiler control, and fuel oil system are practically identical with the U. S. S. *Tennessee* described in the May, 1922, issue of this JOURNAL.

The final official trials of the U. S. S. *Maryland* were held November 14–17, 1921, off Rockland, Maine, and en route Rockland to New York. Just before the trials the ship was docked at the Navy Yard, New York. The ship averaged 21.77 knots on five high speed runs at a displacement of 32,500 tons, developing an average shaft horsepower at this speed of 36,167 with a maximum of 36,544. The trials were most successful and unattended with a casualty of any kind. There is appended below in tabular form the principal data obtained on trials:

STEAMING TRIAL DATA.

	12 Hour Full Power.		4 Hour 19 Knots.		4 Hour 15 Knots.		4 Hour 10 Knots.	
	A	B	A	B	A	B	A	B
Average Speed, knots	21.07	21.07	18.97	18.97	14.98	14.98	10.07	10.07
Displacement, tons	32,655	32,655	32,550	32,550	32,615	32,615	32,518	32,518
Average S.H.P.	29,200	29,750	19,000	19,470	9190	9225	2575	2560
Total oil consumption per hour, pounds, (19,500 B.T.U.)	28,651.6	28,710.5	20,857.9	20,115.9	10,986.1	11,223.7	4912.8	4524.3
Oil consumption per S.H.P., pounds, per hour	.982	.966	1.064	1.002	1.195	1.216	1.902	1.762
Oil consumed per square feet H.S., pounds, per hour	.614	.625	.518	.500	.477	.484	0.427	0.393
Oil consumption per knot, pounds (19,500)	.....	1339	.....	1022	733.9	748.7	.....	446
Water consumption, pounds per S.H.P. per hour	10.817	11.555	10.775	.....	10.915	11.066	13.20	14.56
Number of boilers used during run	8	8	7	7	4	4	2	2
Steam at boilers, pounds gage	250	250	260	260	265	265	250	250
Steam at turbine, pounds gage	246	246	255	255	258.8	260	241.3	243.0
Steam at turbine, temperature degrees F	436.7	436	.....	.....	.....	.....	424	424
Vacuum, inches	29.0	29.0	28.5	28.5	28.0	28.0	28.0	28.0
Barometer, inches	29.98	29.98	30.22	30.22	30.22	30.22	30.20	30.20
Fireroom, air pressure, inches of water	5.5	5.5	4.8	4.8	3.7	3.7	2.8	2.8
Shaft R.P.M.	171.24	171.24	150.74	150.74	118.01	118.01	78.01	78.01

Column A—Auxiliary exhaust into dynamo condenser.
Column B—Auxiliary exhaust into main turbines.

VENTILATION OF SHIPS, WITH SPECIAL CONSIDERATION TO AIR SUPPLY OF FIREROOMS.

By Ing. Julius Lévai.

Translated and Rewritten by Commander O. L. Cox, U. S. Navy, Member.

Adequate and reliable ventilation is most important in the mining industry; the design of fans for this purpose is well developed and a large amount of experimental and actual data of results obtained is available. In order to compare various types and installations, it is desirable to reduce all to a standard basis which has been done by the introduction of the well known principle of equivalent cross section area.

It is the object of this paper to apply that method of observation to the ventilation of ships and especially to fire rooms.

In figure 1, air is supplied through opening S at constant temperature and pressure to space R, and the same amount of air is continuously flowing out through orifice A. The pressure within space R is measured by means of a manometer. Assuming that orifice A is frictionless, then the velocity of air leaving will be:

$$v = \sqrt{2gh} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Where h = pressure in space R, expressed in length (meters or feet) of a column of air in its actual condition.

v = velocity per second (meters or feet).

g = acceleration due to gravity (9.806 or 32.17).

Usually the air pressure is measured by means of a water column or other fluid and in order to determine the hydraulic head h, it is necessary to know the specific gravity of the liquid used in the manometer tube relative to air.

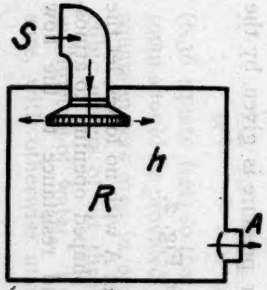


FIG. 1.

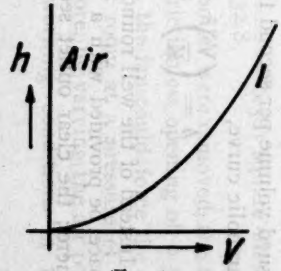


FIG. 2.

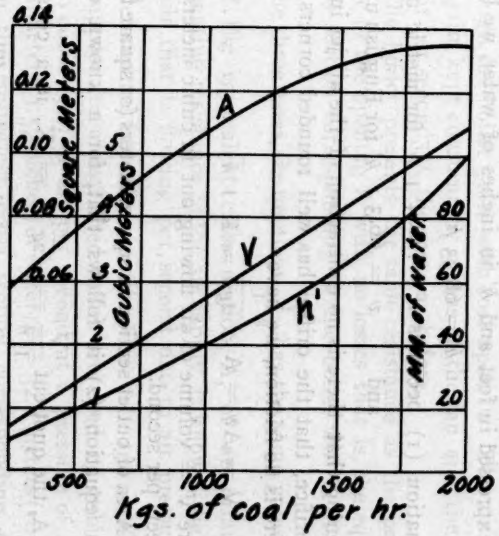


FIG. 3.

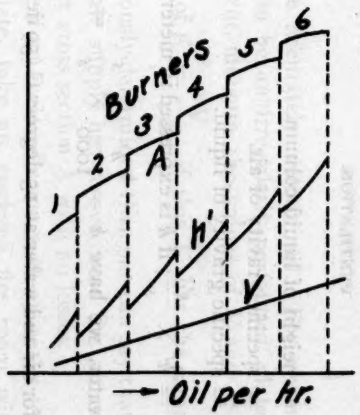


FIG. 4.

Let h' = height of liquid column.

s = specific gravity of air.

s' = specific gravity of liquid.

then $h = h' \times \frac{s'}{s}$; if h is expressed in meters and h' in millimeters water, we have $h = \frac{h'}{1000} \times \frac{s'}{s} = 0.82\ h'$;

$\left[\frac{s'}{s} = 820 \text{ for air and water at 15 degrees C (59 degrees F)}\right]$

If h is expressed in feet and h' in inches of water, we have

$$h = 68.33\ h'$$

and equation (1) becomes $v = 4.04\ \sqrt{h'}$ for metric units

and $v = 66.3\ \sqrt{h'}$ for English units.

Assuming that there is no contraction of the air jet in the outlet orifice, that the orifice has well rounded corners and that there is no friction, we have

$$V = Av = A\ \sqrt{2gh} = K\ \sqrt{h} \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Where V = volume of air flowing out in cubic meters (or cubic feet) per second.

A = Area of outlet section in square meters (or square feet).

From equation (2) it follows that, for a known cross section A, the quotient $\frac{V}{\sqrt{h}} = \sqrt{2g}\ A = K$, is a constant which depends only on the area of the outlet. Thus for a given cross section A (without losses), the relation of the issued volume per second to the air pressure is given by the parabolic curve,

$$h = \left(\frac{V}{K}\right)^2 \text{ shown in Fig. 2.}$$

Instead of the well rounded outlet A with no losses, let the space be provided with a normal shaped opening to atmosphere; the clear outlet section and resistance to the flow

cannot be accurately determined. For example, take the opening to be an irregularly perforated plate. With this arrangement, measure the values of V and h, and calculate the value of $K = \frac{V}{\sqrt{h}}$; if this K has the same value as the one previously determined then one can say that this arrangement of air supply has to overcome the same resistance as the former cross section A with no losses.

Instead of an enclosed room with an outlet through a perforated plate, take an enclosed fire room with the outlet through the boiler grate with coal upon it. The air pressure is again kept constant by means of the fan and the amount of air per second, V, the same as before.

The boiler presents the same resistance as the perforated plate or the opening A with no losses that is, the equivalent cross section corresponding to the boiler will be A. In all of the three cases, the quotient $\frac{V}{\sqrt{h}}$ has the same value.

If the equivalent cross section A, of an arrangement presenting resistance to the air supply, is known, the amount of air that is flowing per second due to air pressure h can be determined from the equation $V = A\sqrt{2gh} = K\sqrt{h}$.

Suppose for a coal fired boiler we need 15 cubic meters (530 cubic feet) of air per second at a pressure of $h' = 40$ millimeters (1.57 inches) of water; $h = 32.8$ meters (107.6 feet), then the equivalent cross section corresponding to these conditions will be:

$$A = \frac{V}{\sqrt{2gh}} = \frac{15}{\sqrt{2 \times 9.806 \times 32.8}} = 0.591 \text{ square meters}$$

(6.36 square feet) so that, if the boiler were removed, the fire room should be provided with a frictionless opening of 0.591 square meters area in order that the fan could force the 15 cubic meters of air through the room at a pressure of 40 millimeters of water. Furthermore, by varying the air pressure, with otherwise unaltered working conditions of the

boiler, the amount of air will vary in the same proportion whether the boiler is there or merely the equivalent opening.

In all cases, the parabolic curve, figure 2, indicates the relation between the air pressure h, and the quantity of air flowing per second V, without considering by what means the air pressure is kept constant in the fire room.

Figure 3, gives the results on a coal fired torpedo boat boiler. The observed values being coal burned per hour, air pressure h' in millimeters of water and volume of air V in cubic meters per second. From corresponding values of h and V the equivalent cross section area A in square meters has been calculated from the equation $A = \frac{V}{4.04\sqrt{h'}}$

Figure 4 represents the estimated results of an oil fired boiler equipped with 6 burners, there being a distinct curve for h' and A for every combination of burners.

The area of equivalent cross section depends, in general, upon the demand on the capacity of the boiler. If economical combustion is taken for granted, V will vary approximately as a straight line and h will depend on the amount of air required (V), on the resistance offered by the boiler (thickness of coal on grate, etc.) and on the strength of the existing natural draft.

By means of the idea of equivalent cross section, a clear judgment is possible when conditions in the fire room vary. For example, a boiler requires 12 cubic meters (423.8 cubic feet) of air per second at an air pressure corresponding to 190 millimeters (7.48 inches) water pressure. The area of the equivalent cross section is 0.2155 square meters (2.32 square feet).

Suppose that the fire room has two round hatches 600 millimeters (23.62 inches) in diameter and that the hatches are not a tight fit but have a gap 3 millimeters (0.118 inches) all around. The net area of the gaps for passage of air is 0.0113 square meters (0.1216 square feet). Assuming a

velocity coefficient of 0.60 for the gap gives an equivalent area of 0.00678 square meters so that the total equivalent area for the fire room is 0.2155 + 0.00678 = 0.22228 square meters (2.39 square feet). The increase in equivalent area is 3.15 per cent which gives an idea as to the effect of leaks in the fire room.

We have to deal with a similar case if there is a gap between the intake pipe and the fan when a fan without casing is involved. The ratio between the gap and equivalent cross section gives an idea of the losses to be expected.

On account of weight and economy, the resistance of the boiler should be kept low and the equivalent cross section kept as large as possible and this can only be done by reducing the air resistance in the boiler itself, which requires careful consideration in the design of the boiler, in order to give the most favorable shape to those parts which offer the most resistance.

If a centrifugal fan discharges into space R, figure 1, and the outlet opening (with no losses) has the constant value A, then the following considerations are valid as long as the room R is large enough, in relation to the fan, so that all the air is brought to rest—that is, all the velocity head is converted into static head. For such a fan of given type, dimensions and structure, there are invariable relations between the supply, air pressure and number of revolutions.

The important factor is the peripheral speed. Let the fan be rotated at different speeds, then for each given number of revolutions per minute a state of equilibrium or permanent condition can be taken for granted,

$$\frac{\pi dn}{60} = u \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where d = outside diameter of fan (meters or feet).
n = number of revolutions per minute.
u = peripheral velocity per second (meters or feet).

The following relations hold between the quantities h, V and u.

(a) The air pressure is proportional to the square of the peripheral speed. $h = C_1 u^2$ . . . . . (4)

(b) The quantity of air supplied is proportional to the peripheral speed. $V = C'_2 F u = C_2 u$ . . . . . (5)

(c) The effective work to drive the fan is proportional to the cube of the peripheral speed. $P = C'_3 F u^3 = C_3 u^3$ (6)

The constant C_1 is not dependent upon the absolute size of the fan but upon its type; that is, the design of blades; their relative arrangement, number, size and shape; upon the ratio of inner to outer diameter, etc.

The constant C_2 depends upon the same factors as C_1 and in addition upon the area (F) through the blades for flow of air ($C'_2 F = C_2$) and the absolute size of the fan. The constant C_3 depends upon the type and absolute size of the fan ($C'_3 F = C_3$).

From equations 3 to 6, the following relations are established in regard to the number of revolutions:

$$h = K_1 n^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4')$$

$$V = K_2 n \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5')$$

$$P = K_3 n^3 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6')$$

These equations give with sufficient accuracy the relations between h, V and P and number of revolutions n. The curves, figure 5, gives these results for an actual fan.

The air horsepower required to deliver a given quantity of air against a given head is

$$\text{Air H. P.} = \frac{V \times h'}{75}$$

Where V is in cubic meters per second and h' is in millimeters of water. If V is expressed in cubic feet per second and h' in inches of water, this becomes — $\text{Air H.P.} = \frac{V \times h' \times 5.2}{550}$

Therefore if P is the power required to drive the fan the efficiency is

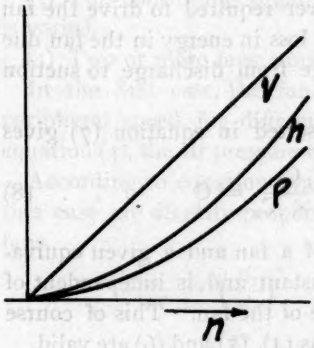


FIG. 5.

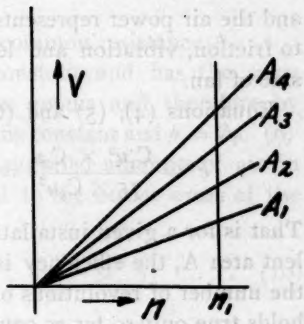


FIG. 6.

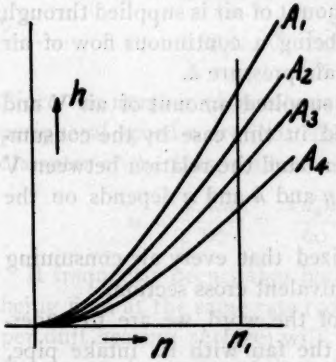


FIG. 7.

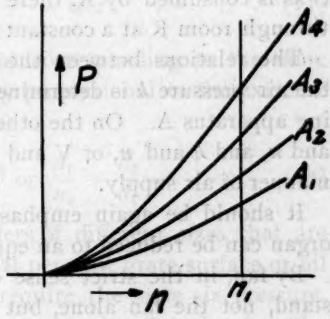


FIG. 8.

$$e = \frac{V \times h'}{75 \times P} \text{ or in English units } e = \frac{V \times h' \times 5.2}{550 \times P} \qquad (7)$$

The difference between the power required to drive the fan and the air power represents the loss in energy in the fan due to friction, vibration and leakage from discharge to suction side of fan.

Equations (4), (5) and (6) inserted in equation (7) gives

$$e = \frac{C_1 u^2 \times C_2 u}{75 \times C_3 u^3} = \frac{C_1 C_2}{75 \times C_3} = C \qquad . \ . \ . \qquad (8)$$

That is for a given installation of a fan and a given equivalent area A, the efficiency is constant and is independent of the number of revolutions or size of the fan. This of course holds true only so far as equations (4), (5) and (6) are valid.

In the arrangement shown in figure 1, S represents the supply and A the consuming organ. The two are to be considered as component parts and are subject to investigation for the case where the same amount of air is supplied through S as is consumed by A, there being a continuous flow of air through room R at a constant air pressure h.

The relations between the supplied amount of air V and the air pressure h is determined in this case by the consuming apparatus A. On the other hand the relation between V and u, and h and u, or V and n and h and n depends on the manner of air supply.

It should be again emphasized that every air consuming organ can be reduced to an equivalent cross section.

By fan, in the strict sense of the word, we are to understand, not the fan alone, but the fan with its intake pipe, together with the structure, casing, the fire room and arrangement of boilers.

With the arrangement as shown in figure 1, four conditions arise:

(1) The fan (S) and consuming organ (A) remain constant and the size of the plant is varied.

(2) The fan and structure remain unaltered and the consuming organ is varied.

(3) The consuming organ remains constant and the fan is varied.

(4) Two or more fans work on common resistance A.

In the first case, the fan is constant and has the same peripheral speed for different size rooms and therefore by equation (4), the air pressure remains constant and $h_2 = h_1$. (9)

According to equation (5), the supplied amounts of air in this case are directly proportional to the outlet areas of the fans.

$$\frac{F_2}{F_1} = \left(\frac{d_2}{d_1}\right)^2 = \alpha^2;\ V_2 = \alpha^2 V_1 \quad . \quad . \quad (10)$$

and finally, $P_1 = \frac{h_1 V_1}{75e}$; $P_2 = \frac{h_2 V_2}{75e}$ where e according to equation (8) is constant and independent of the size of the fan. Substituting from equations (9) and (10), we get

$$P_2 = \frac{h_1 \alpha^2 V_1}{75e} = \alpha^2 P_1 \quad . \quad . \quad . \quad . \quad (11)$$

Since the number of revolutions corresponding to the same peripheral speed u are indirectly proportional to the extreme diameter of the fans, we have

$$u = \frac{\pi d_1 n_1}{60} = \frac{\pi d_2 n_2}{60} \text{ or } \frac{n_1}{n_2} = \frac{d_2}{d_1} = \alpha \quad . \quad . \quad (12)$$

It frequently occurs that boilers of different sizes that are being fired at the same rate (coal per unit grate surface or oil per unit heating surface) will require the same air pressure. By means of equations (9), (10), (11) and (12) and using the dimensions of a satisfactory plant a new plant can be constructed that will give similar results.

Furthermore, the law of similarity renders it possible to determine the characteristics of a type of fan (including the accepted structure) by means of a model plant of diminished scale assuming that the measurements are employed within

such limits that the law of similarity holds for continuity of flow of air, losses of friction, leaky joints, etc.

In figure 1, with a permanent size fan, let the area of resistance be varied; in this case the character of curves "h and n" and "V and n" and "P and n" will remain constant but C_1 C_2 C_3 will have different values. By means of a series of orifice areas A_1 A_2 - - - - A_n, a group of curves of "V and n", "h and n" "P and n", can be laid out.

Figures 6, 7 and 8 represent such curves for an actual installation; taking any given R.P.M. such as n_1, pick out values of V from figure 6, and h from figure 7, corresponding to various outlet areas A_1, A_2 - - - A_n and plotting these values of V and h, we get the curve shown in figure 9 for constant R.P.M.

From the values of P (figure 8) and simultaneous values of V and h (figure 9), the efficiency e can be calculated from equation (7). This is plotted in figure 9. The form of this curve, which is the characteristic of the fan in question, depends on the system of ventilation, that is, form of blades, ratio of outside to inside diameters, the form of intake and discharge pipes, and arrangement of fireroom.

By repeating this procedure for different R.P.M.'s, the group of curves, figure 10, will be obtained showing the relation between amount of air V, air pressure h and R.P.M. for all conditions.

After conducting the experiment and plotting figures 6, 7 and 8 the constants K_1, K_2 and K_3 can be obtained from equations (4′), (5′) and (6′). Likewise C_1, C_2 and C_3 can be obtained from equations 4, 5 and 6. These values are constant for any particular case, yet have different values for each different value of A.

If F is the outlet area of the fan, neglecting the thickness of blades, we have:

$$F = \pi d w \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

where d = outer diameter
w = length of blades

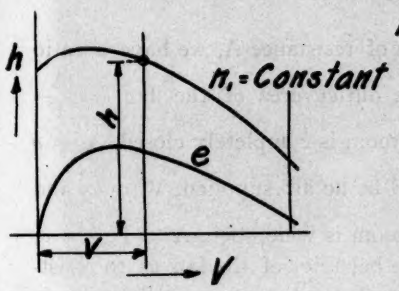


FIG. 9.

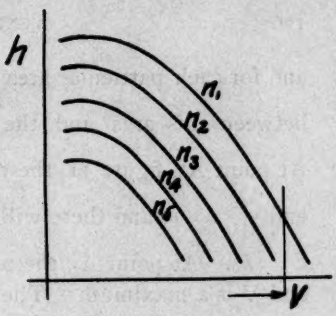


FIG. 10.

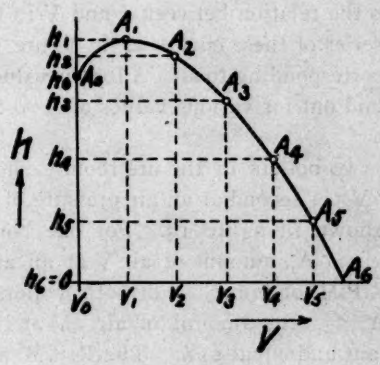


FIG. 11.

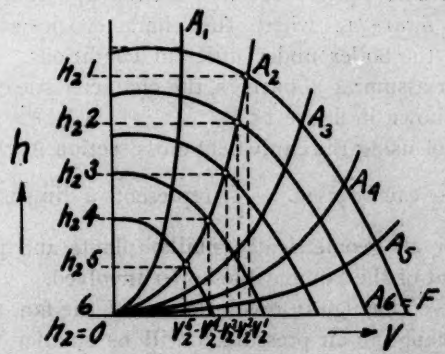


FIG. 12.

and for each particular area of resistance A, we have a ratio between this area and the outlet area of the fan $= \frac{A}{F}$. At point A_o, figure 11, the room is completely closed, $A = o$ and $\frac{A}{F} = o$, and there will be no air supplied, $V = o$, and $h = h_o$. At point A_6, the room is wide open, $A = F$, $h = o$, and V is a maximum. The behavior of the fan as to resistance can be readily followed on this curve. Suppose it works against area of resistance A_1, then the parabolic curve, figure 2, gives the relation between h and V in the room and laying out a series of these curves as in figure 12, we obtain the R.P.M.'s corresponding to area A for the values of h and V. Figure 12 is laid out for various values of A so that it shows all conditions.

If there are two boilers in the fire room each requiring an amount of air V per second at an air pressure of h, the necessary data is shown in figure 13. For one boiler only, the equivalent area is A_1, amount of air V at an air pressure h, which gives R.P.M. of fan n_3. For both boilers, the equivalent area is $A_2 = 2A_1$, amount of air 2V and air pressure remains constant and equal to h. The R.P.M. are now n_1.

When a boiler is being worked at different rates, it represents a variable equivalent cross section and the results are shown in figure 14 where the characteristic curve "H" represents the boiler under different conditions. For an oil fired boiler assuming 6 burners, the characteristic curve takes the form shown in figure 15.

Instead of using the equivalent cross section A, the relation of areas $\frac{A}{F}$ can be used. $\frac{A}{F}$ represents a dimension which is valid for all geometrically similar plants and practically independent of the fan and the boiler involved.

For a given change in revolutions n of the fan, the corresponding change in air pressure h will be smaller the larger the equivalent cross section area A and the same is true if the

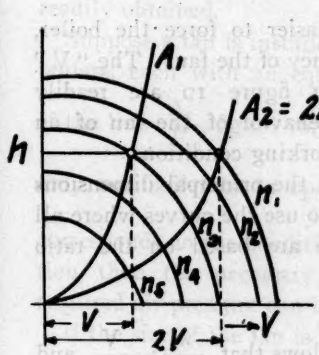


FIG. 13.

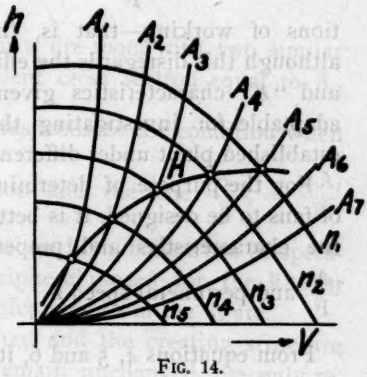


FIG. 14.

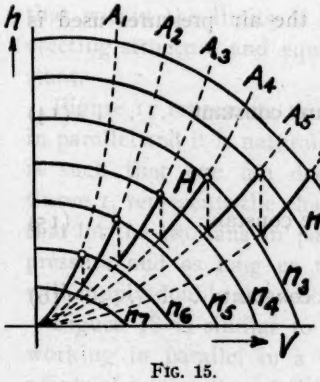


FIG. 15.

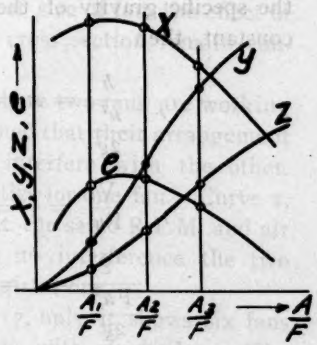


FIG. 16.

ratio $\frac{A}{F}$ is used instead of A. Generally speaking, the smaller the ratio $\frac{A}{F}$, the better the fan for different conditions of working—that is, the easier to force the boiler, although this disregards the efficiency of the fan. The "V" and "h" characteristics given in figure 10 are readily adaptable for investigating the behavior of the fan of an established plant under different working conditions.

For the purpose of determining the principal dimensions of fans to be designed, it is better to use the curves where all the characteristics and properties are based on the ratio $\frac{A}{F}$ and peripheral speed u.

From equations 4, 5 and 6, it follows that $\frac{h}{u^2}$; $\frac{V}{u}$ and $\frac{P}{u^3}$ are constant, and assuming that for practical purposes the specific gravity of the air at the air pressures used is constant, then

$$\frac{\frac{h}{u^2}}{2g} = x = \text{Pressure constant} \quad . \quad . \quad . \quad (14)$$

$$\frac{V}{Fu} = y = \text{Volume constant} \quad . \quad . \quad . \quad (15)$$

$$\frac{\frac{P}{Fu^3}}{2g} = z = \text{Power constant} \quad . \quad . \quad . \quad . \quad (16)$$

are constant and independent of the size of the fan and its R. P. M. and are variable in proportion to the equivalent area A or the ratio $\frac{A}{F}$.

Figure 16 shows the relations between the values of x, y,

s and e and the ratio $\frac{A}{F}$. From this all the principal dimensions and the power requirements of the type of fan can be readily obtained.

Suppose a fan is installed in a fire room with two similar boilers, each with an equivalent cross section equal to A_1 If the line $\frac{A_1}{F}$, figure 16, corresponds to the condition when only one boiler is in operation and the line $\frac{A_2}{F} = \frac{2A_1}{F}$ corresponds to the condition when both boilers are in operation, then the necessary peripheral speed for the known required air pressure can be calculated from the figure.

If the size of the fan is altered and the erecting structure and equivalent cross section remain unchanged, the only result will be a change in value of $\frac{A}{F}$. This of course assumes that within the limits of change of size of fan, the effect of erecting structure and equivalent cross section remain constant.

Figure 17 represents the case where two fans are working in parallel and it is naturally assumed that their arrangement is such that one fan does not interfere with the other. Curve 1, represents the characteristics for one fan. Curve 2, that for the two fans in parallel at the same R.P.M. and air pressure and as long as there is no interference the two will supply double the amount of air as one.

Figure 18 is similar to figure 17, only it shows six fans working in parallel in a fire room with six boilers. The points of intersection of the curves of equivalent cross section A, with the characteristic curves determine the amount of air and the air pressure required.

Figure 18 shows that the conditions might be such that a few fans at the same R.P.M. would give more air and greater air pressure than a larger number. Take the inter-

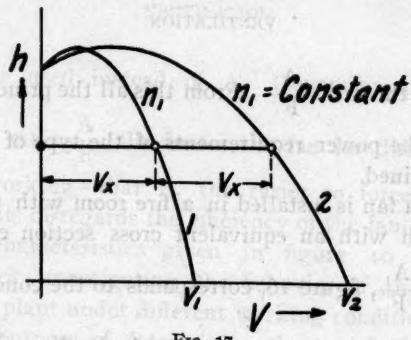


FIG. 17.

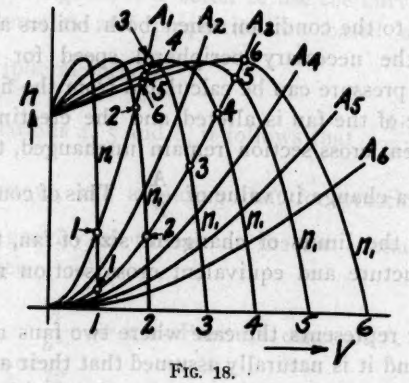


FIG. 18.

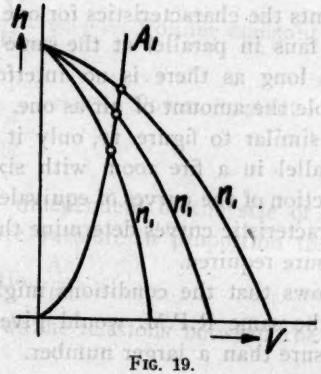


FIG. 19.

section of the curve A_1 with the characteristic curves. This shows that the maximum air pressure and maximum volume of air at R.P.M. of n_1 is obtained by using three fans.

As the equivalent area is increased A_1 - - - - A_6 it will be seen that this unfavorable condition no longer holds and there is a point beyond which increasing the number of fans increases the amount of air and the air pressure. The above unfavorable condition will only arise with fans whose blades are curved forward in the direction of rotation. For fans whose blades are curved backwards, the curve of characteristics is a rapidly descending one as shown in figure 19.

The following shows the procedure to determine the equivalent cross section of a boiler of a ship by means of observations on a trial trip.

The equivalent cross section of the boiler and fire room is $A = \frac{V}{\sqrt{2gh}}$ and this area remains constant for the different temperatures and pressures of the air and in order to calculate the value of A, the amount of air V being forced through the fire room and the air pressure h must be measured under steady working conditions. The amount of air V, can be determined in the following manner:

$$V = \frac{B \times L \times r}{3600} \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

where B = Amount of fuel used per hour

L = Theoretical amount of air of combustion per unit weight of fuel

r = Ratio of actual to theoretical amount of air

From an analysis of the fuel, we obtain the percentage of carbon, hydrogen and oxygen. Let C, H and O represent these percentages and neglecting sulphur we obtain for the theoretical amount of air.

$$L = \frac{22.4}{0.21}\left(\frac{C}{12} + \frac{H}{4} - \frac{O}{32}\right) \quad . \quad . \quad . \quad (18)$$

cubic meters per kilogram of dry coal. To reduce this to cubic feet per pound of coal, multiply equation (18) by the conversion factor 16.03.

Under normal operating conditions a certain amount of carbon is not consumed and is wasted in the ash. Let C_a equal actual amount of carbon consumed, then equation (18) may be written—

$$L = \frac{22.4}{0.21}\left(\frac{H}{4} - \frac{O}{32}\right) + 8.89\,C_a \quad . \quad . \quad (19)$$

For fuel oil the equation for theoretical amount of air becomes

$$L = \frac{22.4}{0.21}\left(\frac{C}{12} + \frac{H}{4}\right) \quad . \quad . \quad . \quad (20)$$

If an oil of which C = 88 per cent and H = 12 per cent is used, this gives L = 11 cubic meters of normal air per kilogram of oil = 176 cubic feet of air per pound of oil.

If the analysis of the coal is known a diagram such as figure 20 can be prepared from which, with the actual percentage of refuse, the amount of air per unit weight can be readily obtained.

Figure 20 is a diagram for coal of the following analysis:

Ash	—	5.10 per cent.
Carbon	—	87.30 per cent.
Hydrogen	—	4.40 per cent.
Sulphur	—	0.90 per cent.
Oxygen	—	2.30 per cent.

By analyzing the smoke pipe gases and obtaining the volume percentage of CO_2, CO, O and N, the amount of surplus air r can be obtained.

$$r = \frac{21N}{21N - 79\,(O - \frac{1}{2}\,CO)} = \frac{21}{21 - 79\left(\frac{O - \frac{1}{2}\,CO}{N}\right)} \quad (22)$$

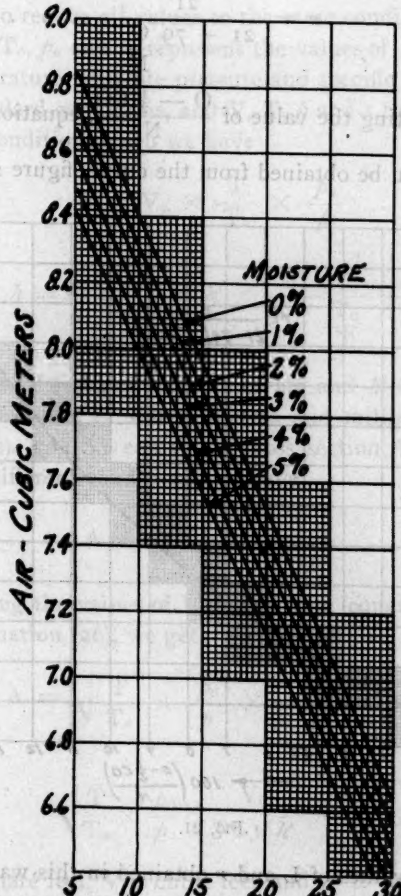

9.0
8.8
8.6
8.4
8.2
8.0
7.8
7.6
7.4
7.2
7.0
6.8
6.6
AIR - CUBIC METERS
MOISTURE
0%
1%
2%
3%
4%
5%
5
10
15
20
25
30
REFUSE (%)

FIG. 20.

and for complete combustion this becomes—

$$r = \frac{21}{21 - 79\,\frac{O}{N}} \quad . \quad . \quad . \quad . \quad . \quad (23)$$

By calculating the value of $\frac{O - \frac{1}{2}\,CO}{N}$, equation (22), the value of r can be obtained from the curve, figure 21.

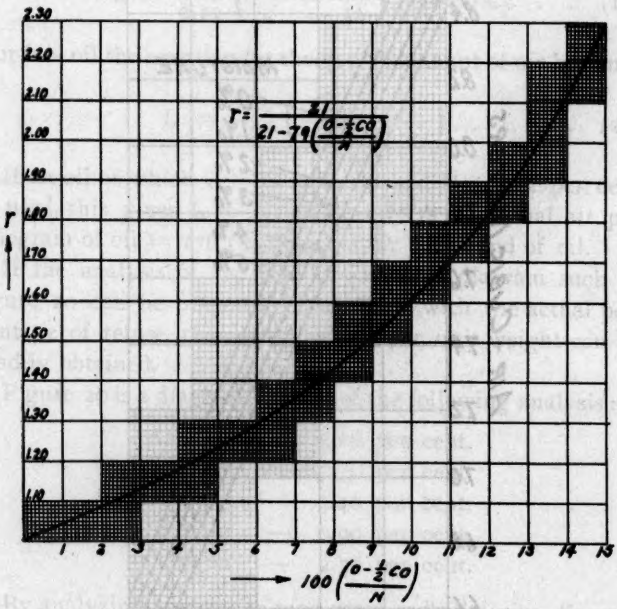


FIG. 21.

From the values of L and r obtained in this way and with the total amount of coal used per hour as measured, the amount of air V forced through the boiler per second can be ascertained from equation (17). This value of V is for air under normal conditions of O degrees C. and pressure of 760 millimeters. The air being measured is at a certain temper-

ature t degrees C. and under a certain absolute pressure p, and in order to obtain the equivalent cross section it will be necessary to reduce all values to the same condition.

Let V_o, T_o, p_o and s_o represent the values of volume, absolute temperature, absolute pressure and specific gravity of air under standard conditions, and V, T, p and s those for actual working conditions, then we have

$$V = V_o \times \frac{T}{T_o} \times \frac{p_o}{p} \quad . \quad . \quad . \quad . \quad (24)$$

and

$$h = h' \times \frac{1}{s} = h' \times \frac{1}{s_o \times \frac{T_o}{T} \times \frac{p}{p_o}} \quad . \quad . \quad (25)$$

Where h = height of column of air and h' the equivalent height of column of water (meters and millimeters or feet and inches). As the equivalent cross section A remains constant for different conditions, we have

$$A = \frac{V_o}{\sqrt{2gh_o}} = \frac{V}{\sqrt{2gh}} \quad . \quad . \quad . \quad . \quad . \quad (26)$$

Substituting the values of V and h from equations (24) and (25) in equation (26), we get:

$$A = \sqrt{\frac{T}{T_o} \times \frac{p_o}{p}} \times \frac{V}{\sqrt{\frac{2g}{s_o}}\sqrt{h'}} =$$

$$\sqrt{\frac{T}{T_o} \times \frac{p_o}{p}} \times \frac{V_o}{3.9\sqrt{h'}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (27)$$

If A is square feet, V is cubic feet and h' is inches of water, this becomes:

$$A = \sqrt{\frac{T}{T_o} \times \frac{p_o}{p}} \times \frac{V_o}{64.4\sqrt{h'}} \quad . \quad . \quad . \quad (27a)$$

Finally if the pressure ratio $\frac{p_o}{p}$ is considered equal to unity equation (27) becomes:

$$A = \frac{V_o}{B\sqrt{h}} \quad (28) \text{ where } B = 3{\cdot}9\sqrt{\frac{T_o}{T}}$$

The values of B for temperatures from O degrees C. to 35 degrees C. are given by the curve, figure 22.

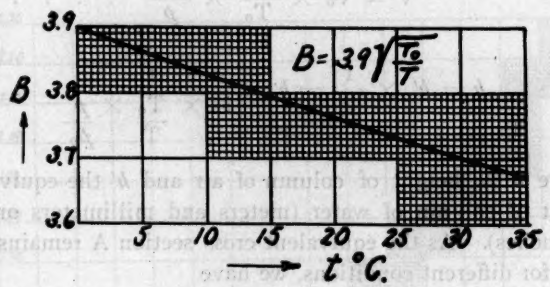


FIG. 22.

With artificial draught and enclosed fire room, V_o represents with sufficient accuracy the amount of air required so that it can be used for estimating the capacity of the fan.

NOTES ON INSPECTION AND MANUFACTURE OF STEEL CASTINGS FOR NAVAL SHIPS.

By Commander Carlos Bean, U. S. Navy, Member.

In August, 1921, the Fore River Plant of the Bethlehem Shipbuilding Corporation, at the suggestion of the Naval Inspector of Machinery, adopted the practice of pickling all Engineering steel castings for both their Naval and their Commercial work. This procedure was suggested for two reasons: First, pickling loosens the scale and sand that adhere to the casting so that it can be perfectly cleaned by wire brushing. Sediment in the lubricating oil system and steam lines is thereby practically eliminated, and, as a result there are very few scored valve seats to grind in, and the lubricating oil lines are free from obstruction. Second, pickling will search out and render visible, defects in a casting which would not otherwise be known until disclosed by the hydrostatic test when considerable time and labor, *i. e.*, money had been spent on it. Or yet, a casting which has not been pickled might pass the prescribed hydrostatic and steam tests only to fail in service, because of invisible defects. This statement is made because during the last four years the writer has inspected many hundreds of castings, many of which appeared perfect as delivered from the foundry, but which, after pickling, developed such fatal defects that they were rejected without question. It is possible that some of these defective castings could have passed the Inspection tests, but would have subsequently failed in service. Pickling brings out these defects and removes a positive source of danger.

Rejection of a casting at a shipbuilding plant unavoidably causes a financial loss, even though replacement is made without charge, due to freight, handling and pickling charges. Additional losses are involved in cases where defects which demand rejection are not discovered until after the necessary machining of flanges for the hydrostatic test has been done. A greater loss is incurred, however, when satisfactory castings can not be secured in time to permit the orderly progress of the machinery installation to continue; tests of main and auxiliary machinery are often held up and, consequently, completion of the ship is delayed. Inability to secure satisfactory steel castings has been the subject of many claims for delays by shipbuilders.

The two defects which cause rejection of castings are: (1) Porosity, (2) Cracks. The percentage of rejections due to porosity is relatively small compared to that due to cracks. Porosity, unless general in a casting, can often be repaired. The most satisfactory method of repair is to drill out the porous spot, tap the hole and fit in a screwed plug, then rivet over the ends of the plug. Never drill out a hole more than one inch in diameter—drill two holes, if necessary, to keep within the dimension.

One manufacturer, for a period of over four months, has been able to supply steel castings to the Bethlehem Shipbuilding Corporation that have been consistently free from porosity. Investigation reveals the fact that he never uses scrap having over .04 per cent of either sulphur or phosphorous. It was also noted that he lays particular stress on the necessity of keeping the metal in the furnace for at least fifteen minutes, after it has become molten, in order to allow gases to escape and give the molten mass a chance to become homogeneous. Another important point is to allow the molten metal to remain in the ladle for at least ten minutes before finally pouring the mold in order to allow any remaining entrained gases to escape.

In discussing the subject of cracks in steel castings, some idea of the seriousness of this defect is given by the following observation:

(1) 50 per cent of the steel castings for recent Naval construction submitted for inspection have been rejected.

(2) 90 per cent of these rejections have been on account of cracks.

(3) 90 per cent of the castings accepted have shown cracks of small depth and length.

When it comes to a question of acceptance or rejection of a casting which has some defects, which may or may not be dangerous, the following procedure is suggested: Examine the casting carefully inside and out under a good light; see if cracks which appear on the outside extend entirely through; if there is no evidence of a complete fracture, drill a hole along the fracture and determine the depth—do not attempt to chip as the metal flows to a certain extent under the blows of a chipping hammer and obliterates trace of the fracture. See if these cracks extend around the casting. If they do and are of any considerable depth, the casting should be condemned. These cracks most often appear around bends and in fillets in portions of the casting subject to alternate contraction and expansion, as steam is turned off or on. The accordion motion set up by alternate strains and stresses would certainly tend to weaken the casting and finally cause a rupture. A longitudinal crack, not at a bend nor in a fillet, is generally shallow in depth and does not impair the strength of the casting, but, in no case should welding of a crack be allowed, because a weld in a location subject to alternate contraction and expansion would not add to the strength or safety of the casting. The extent and depth of the defect should be determined by drilling, as these cracks are very deceptive, and, as shown by experience, an apparent surface check may be deep and seriously impair the vital strength of a casting.

It will be noted from the preceding that the writer has purposely refrained from referring to these defects as "shrinkage cracks," the reason being that cracks in castings are due to lack of provision being made both for expansion and for contraction, and consequently there are "expansion" as well as "shrinkage" cracks. The manufacturers of steel castings have not taken into consideration the fact that molten steel expands immediately before hardening. The heavier part of the casting naturally cools and hardens more slowly, does not expand with the lighter part, and being softer, cracks at the juncture of the heavier and lighter parts or in fillets or bends unless provision has been made for freedom for expansion. Shortly after the expansion takes place, there is a sharp contraction, which will cause the casting to crack unless provision is made to allow for shrinkage.

A careful study of castings submitted for inspection revealed the fact that where the castings could shrink away from the cores there was no sign of contraction cracks, but where the castings shrank on to the cores contraction cracks were numerous.

Various methods of allowing for the necessary expansion may be employed. A complicated casting would require special provision. Ordinarily a layer of coke, or sawdust, or a mixture of these two materials with sand, about three inches thick, separated from the casting by about three inches of molding sand, will compress readily enough to allow for this expansion. Shrinkage is provided for by using collapsible cores or cores that compress easily. In fact, any method that will allow the casting to contract without gripping the core would be satisfactory.

Before this principle of allowing for contraction and expansion was employed 75 per cent of all the steel castings received by the Bethlehem Shipbuilding Corporation from a local steel casting company were rejected on account of cracks. Since this method has been used by the same foundry, not one out of a total of thirty similar castings has been rejected.

The foregoing observations are made as the result of actual experience and apply primarily to medium and large size valve castings and steam fittings, but no doubt apply equally to all steel castings. Practically perfect steel castings are now being made by two steel casting companies who have applied these principles the writer has endeavored to explain. Better methods will be evolved, better castings will result and the losses in time and money will no longer be associated with the use of steel castings.

After careful study and experiments the following methods of allowing for contraction and expansion were suggested, have been tried out and found satisfactory.

METHODS USED BY THE ROXBURY STEEL CASTINGS COMPANY IN CASTING STEEL VALVE BODY FOR U. S. SCOUT CRUISERS NOS. 7 AND 8.

Flask: A metal flask with portable sides is employed. These sides A & B, Figure No. 1, are secured by dogs in such a manner that they may be quickly removed, thus exposing ends C & D of cores.

Core: The core shown in plan in Figure No. 1, and section, Figure No. 2, is made up in sections as follows:—Main body E, tapered plugs F and G, and loose pieces H and J. The portion K is made in the form of a shell core with reinforcing wires inserted as shown in Figure No. 3. Each of these reinforcing wires covers only a portion of the circumference of the shell, thus giving sufficient support to the core while the metal is being poured, but allowing the shell to yield when shrinkage of the metal takes place.

Mold: A large percentage of the defects in castings of these valve bodies occurs at points L, where a large volume of metal is concentrated and remains in a plastic state while the thinner portions cool, contract and pull away from the warmer sections, thus causing defects at the heavier sections. To provide

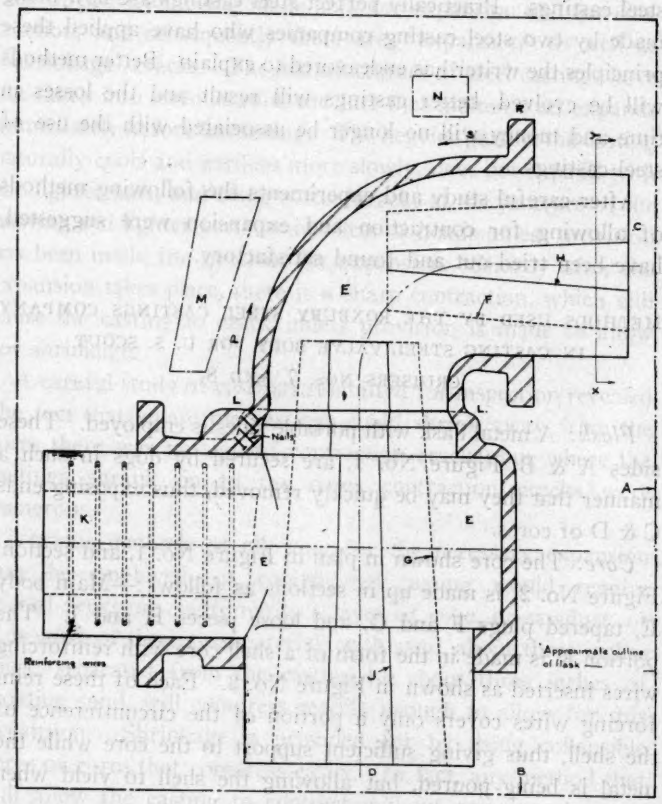

N
R
S
X
C
H
M
E
F
Q
P
L
Nails
K
A
G
Reinforcing wires
Approximate outline of flask
J
D
B

FIG. I.

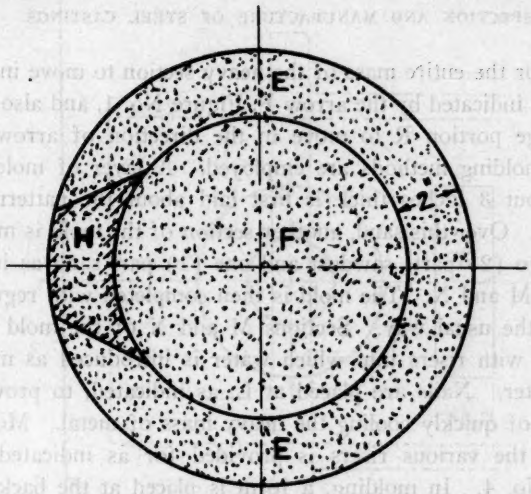


FIG. 2.

SECTION OF CORE AT X-Y.

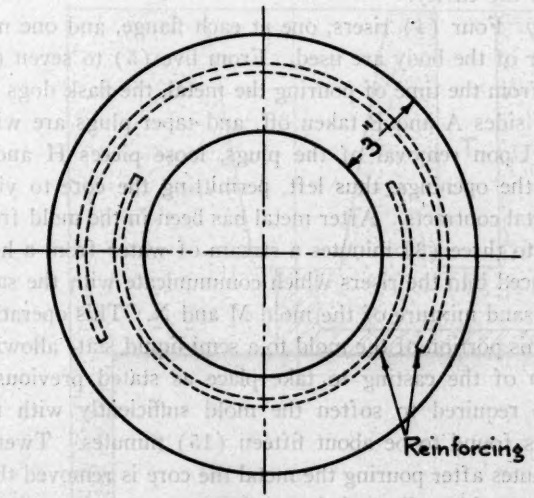


FIG. 3.

ELEVATION OF END K OF CORE.

means for the entire mass of the heavy section to move in the direction indicated by the arrow P, Figure No. 1, and also for the flange portion R to move in the direction of arrow S, special molding methods are employed. A layer of molding sand about 3 inches thick is first laid about the pattern at R and L. Over this sand, another section of the mold is made up of two (2) parts sawdust and one (1) part sand as indicated at M and N. The mold is then completed with regular sand in the usual way. Sections M and N of the mold are provided with risers into which water is introduced as mentioned later. Nails are placed at L, as indicated, to provide a means of quickly cooling the heavy mass of metal. Movement at the various risers is provided for as indicated in Figure No. 4. In molding, a form is placed at the back of each riser as shown at T, Figure No. 4. After completing the mold, the forms are removed and the cavity T, thus formed, is filled with sawdust to prevent the sand from falling back into the cavity.

Casting: Four (4) risers, one at each flange, and one near the center of the body are used. From five (5) to seven (7) minutes from the time of pouring the metal, the flask dogs are removed, sides A and B taken off, and taper plugs are withdrawn. Upon removal of the plugs, loose pieces H and J fall into the openings, thus left, permitting the core to yield as the metal contracts. After metal has been in the mold from two (2) to three (3) minutes a stream of water from a hose is introduced into the risers which communicate with the sawdust and sand mixture of the mold M and N. This operation reduces this portion of the mold to a semi-liquid state allowing expansion of the casting to take place as stated previously. The time required to soften the mold sufficiently with the water was found to be about fifteen (15) minutes. Twenty (20) minutes after pouring the metal the core is removed thus permitting rapid cooling of the casting.

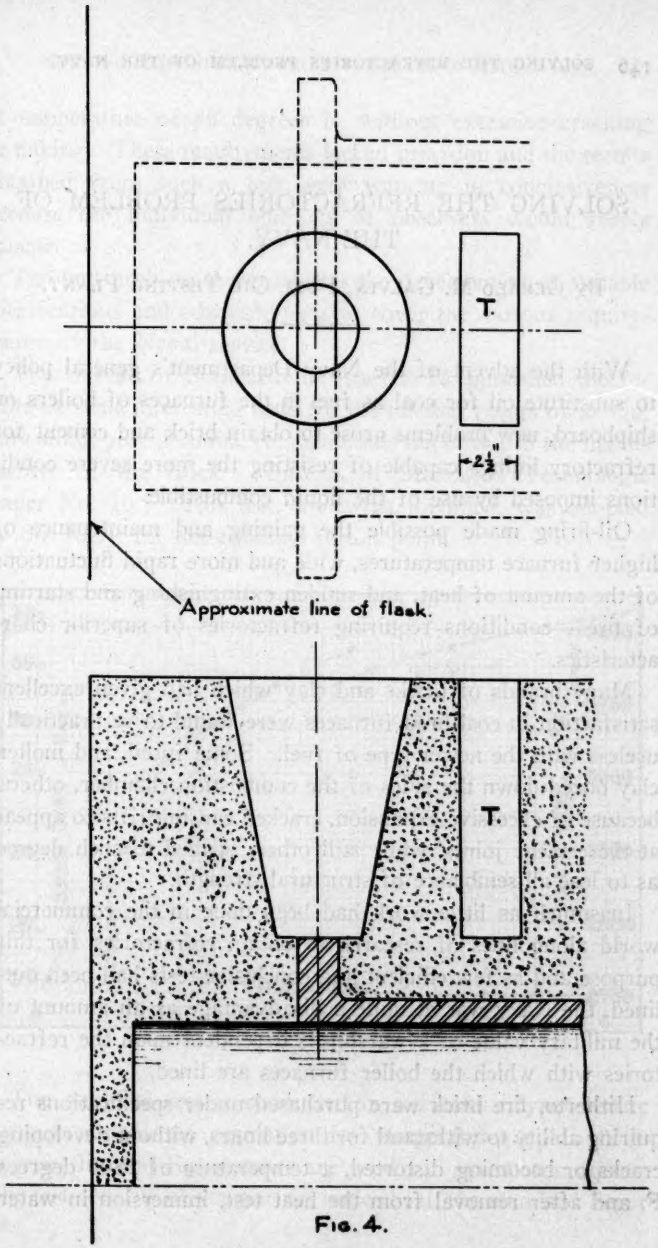


FIG. 4.

PLAN AND SECTION AT RISERS.

SOLVING THE REFRACTORIES PROBLEM OF THE NAVY.

By Gerald M. Galvin, Fuel Oil Testing Plant.

With the advent of the Navy Department's general policy to substitute oil for coal as fuel in the furnaces of boilers on shipboard, new problems arose to obtain brick and cement for refractory linings capable of resisting the more severe conditions imposed by use of the liquid combustible.

Oil-firing made possible the gaining and maintenance of higher furnace temperatures, wide and more rapid fluctuations of the amount of heat, and sudden extinguishing and starting of fires—conditions requiring refractories of superior characteristics.

Many brands of bricks and clay which had given excellent satisfaction in coal-fired furnaces were found to be practically useless with the newer type of fuel. Some fused, and molten clay oozed down the sides of the combustion chamber, others, because of excessive expansion, cracked causing rifts to appear at the cement joints, while still others spalled to such degree as to lose all semblance of structural strength.

Inasmuch as little work had been done in the commercial world along lines of research to obtain refractories for this purpose and as few efficient and conclusive tests had been outlined, the Navy was at quite a disadvantage as an amount of the military value of a warship is dependent upon the refractories with which the boiler furnaces are lined.

Hitherto, fire brick were purchased under specifications requiring ability to withstand for three hours, without developing cracks or becoming distorted, a temperature of 2800 degrees F. and after removal from the heat test, immersion in water

at temperature of 80 degrees F. without excessive cracking or flaking. These requirements lacked precision and the results obtained from such a test were wanting in conclusiveness because the individual opinions of observers would rarely concur.

The first problem therefore was the development of suitable specifications and adequate tests to cover the various requirements of the Naval service.

The Bureau of Standards in June, 1912, published the results of some interesting tests for the melting points of kaolin, pure silica, pure alumina, etc., materials important in the manufacture of fire brick. (Bureau of Standards Technologic Paper No. 10.) This test, somewhat modified, was adopted by the Navy for determining fusion point.

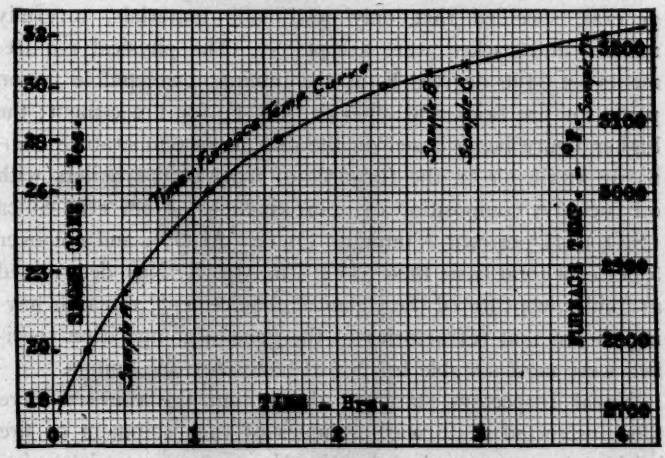


FIG. 1.

The modified test is conducted in a Hoskins type electric furnace, capable of regulation for temperatures up to 3500 degrees F., and Seger cones are used for pyrometric agents. The test consists of comparing, under heat, the standard Seger

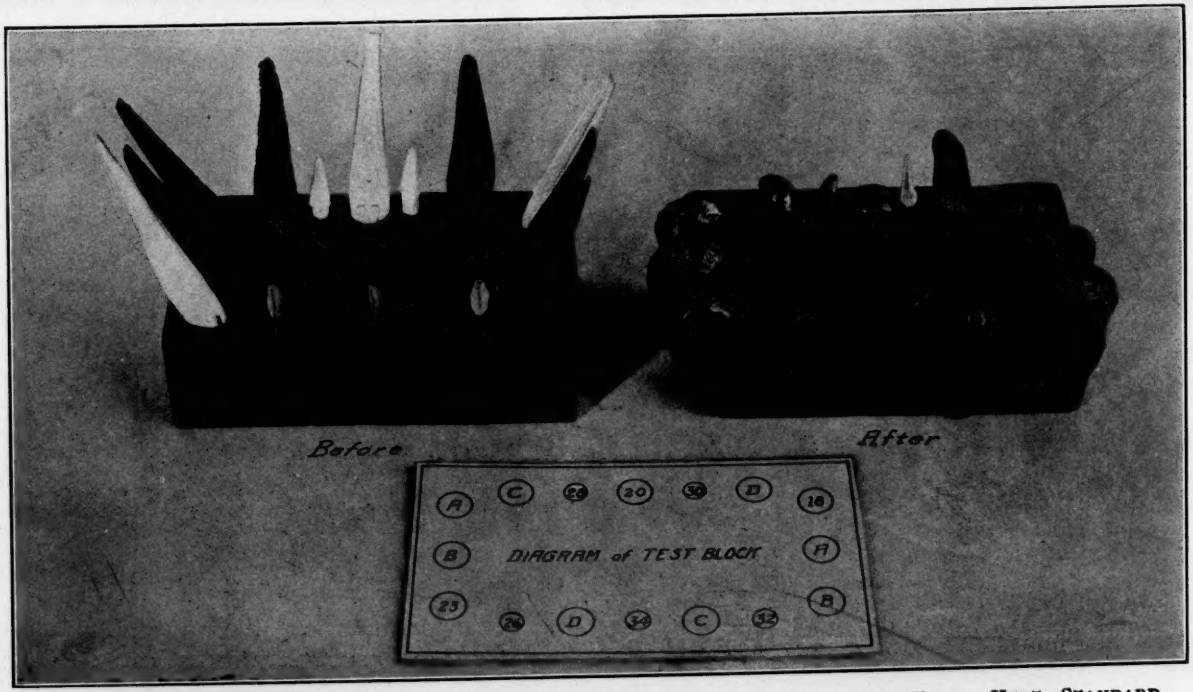
cones with similar cones moulded of the material under examination. Test cones of brick are obtained by pulverizing fragments broken from the corners of a quartered brick; sufficient dextrine and water is added to act as a binder and the resulting mixture pressed into molds. After thoroughly drying, these cones with Seger cones are placed in the furnace and observations recorded of the time when each softens. A curve, Fig. 1, is plotted from the data obtained.

Examination was made of hundreds of brands of refractories on the market; this research furnished data for Navy Department Specifications No. 50B6 issued May 1, 1916.

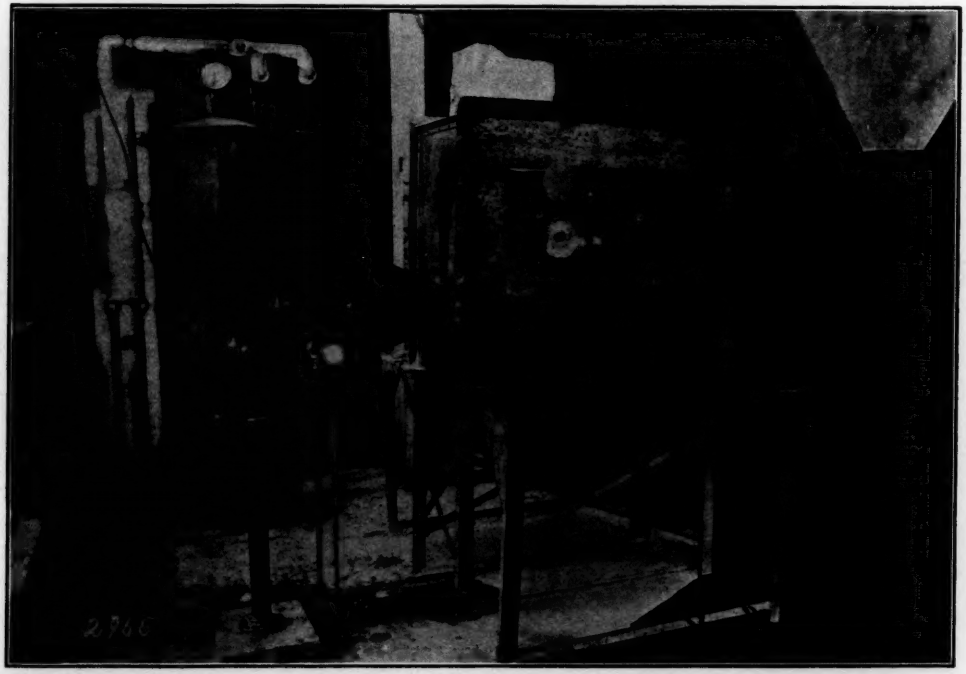
SPECIFICATIONS FOR REFRACTORIES.

These specifications embodied workmanship, physical and chemical requirements. The workmanship requirements, in general, specified that only well-made materials, carefully marked and packed were acceptable. The physical requirements set 3100 degrees F. as the minimum acceptable fusion point for fire brick and 3000 degrees F. for fire cement; the higher figure being based on the maximum temperature ordinarily attainable in the furnaces of marine type boilers with oil fuel. A composition of approximately 54 per cent Silica (SiO_2), 41 per cent Aluminum Oxide (Al_2O_3), and not over 5 per cent total basic fluxes (Fe, Ca, Mg, K, Na) determined as oxides, constituted the chemical requirement; research having shown that refractories of the general composition specified were most suitable for Naval boiler furnace linings.

Naval Inspectors stationed throughout the country were then instructed to forward samples of each shipment of fire brick and cement purchased under Navy contract to determine conformity of material with requirements. After noting general characteristics and workmanship, the acceptance test consisted of chemical analysis and determination of softening temperature; the latter was considered of major importance and was the test on which final acceptance was based.



SOFTENING TEMPERATURE IS DETERMINED IN AN ELECTRIC FURNACE BY COMPARING, UNDER HEAT, STANDARD PYROMETRIC CONES WITH SIMILAR CONES MOULDED OF THE MATERIAL TO BE TESTED. NUMERALS ARE PYROMETRIC CONE NUMBERS; SAMPLES ARE INDICATED ALPHABETICALLY.



FURNACE IN WHICH SIMULATIVE SERVICE TESTS ARE CONDUCTED.

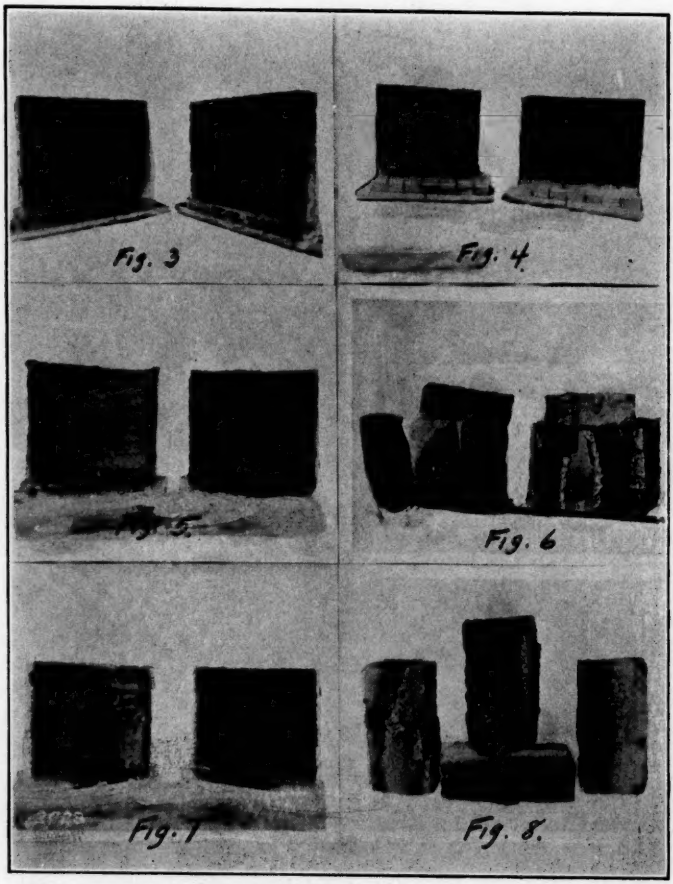


Fig. 3
Fig. 4
Fig. 5.
Fig. 6
Fig. 7
Fig. 8.

It was later considered desirable to adopt some method of determining, under conditions similar to those encountered in service, the relative merits of brands which had passed the acceptance test. This together with broad claims of super-refractoriness made by manufacturers of special brick and cement, made necessary the development of a comparative test.

Lining boiler furnaces with the material under examination was resorted to but this method was both slow and expensive. In the latter part of 1918, a simulative service test was developed at the Fuel Oil Testing Plant.

THE SIMULATIVE SERVICE TEST.

This test is conducted in a small oil-fired furnace, Fig. 2. One side wall of the combustion chamber is constructed of brick and cement of the brands submitted for test, the opposite side wall of brick and cement of brands which have proven satisfactory under test and service usage. A compressed-air-atomizing fuel oil burner is used. The flame sweeps the length of the furnace, returns, and escapes through stack outlet. The test consists of three "runs" at temperatures of 2800, 2900, and 3000 degrees F., respectively, each of 24 hours duration. During the "runs" furnace temperatures are ascertained at quarter-hourly intervals with a Morse Type Optical Pyrometer sighting into the furnace above the burner. To compare heat-insulating properties of the walls, temperatures of the outer faces are determined half-hourly with the optical pyrometer sighting on the brickwork through sillimanite tubes, the ends of which are placed flush with the wall. Immediately after each "run," a spalling test is conducted by injecting cold air at high velocity into the furnace until the walls are cool. The results of the test are obtained by comparison; the comparative heat-insulating properties together with the relative conditions of the competing sidewalls at the conclusion of the test determining whether or not the material under test is acceptable.

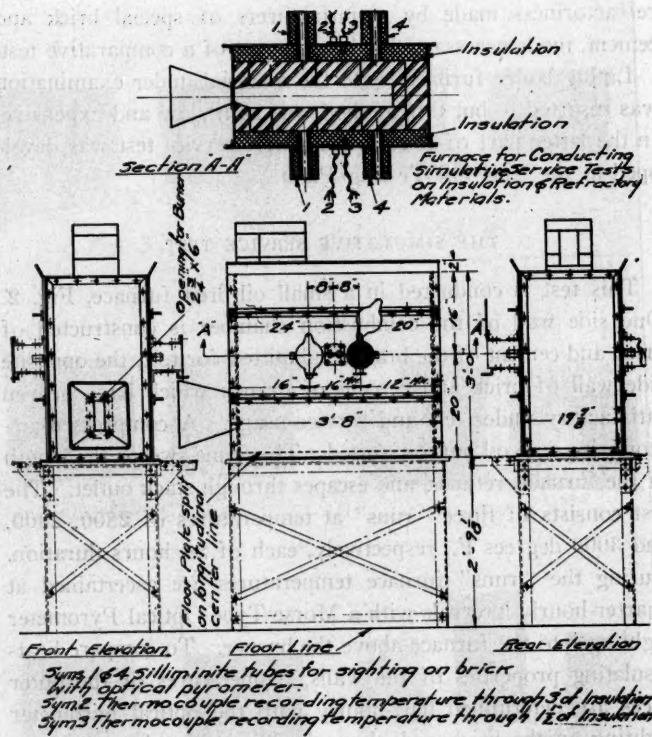

Insulation
Insulation
Section A-A
Furnace for Conducting Simulative Service Tests on Insulation & Refractory Materials.
Opening for Burner 2½ x 6
Floor Plate split on longitudinal center line
Front Elevation.
Floor Line
Rear Elevation
Sym1. 1/64 Sillimanite tubes for sighting on brick with optical pyrometer.
Sym2. Thermocouple recording temperature through 3" of Insulation
Sym3. Thermocouple recording temperature through 1½" of Insulation

FIG. 2.

RESULTS OF TESTS.

The simulative service test has proven an excellent criterion, presenting at its conclusion many varied conditions of brickwork. Some brands have shown themselves to be "true oil-bricks," Fig. 3. Badly spalled brickwork is shown on Fig. 4. The effect of a poor quality brick and high temperature cement may be noted on photograph, Fig. 5. Here the high temperature cement with which the bricks were laid obviously possessed greater heat resistance as the joints are clearly discernible protruding beyond the brickwork. Fig. 6 is a photograph of units of walls shown on Fig. 5 in which the depth of melting action can be gauged. The opposite of Fig. 5 is apparent in Fig. 7, when during the first run at a furnace temperature of 2800 degrees F., melted cement oozed from the joints permitting the bricks to slide from position, exposing them, with disastrous results, to the direct action of the flame. The first run also disclosed the undesirable laminations shown in the photograph, Fig. 8.

METHODS NOW USED.

The present practice of the Navy is to purchase refractories under annual contracts awarded only for brands that have proven satisfactory under test, the names of which are registered on the Department's list of acceptable fire brick and fire cement manufacturers.

To stimulate competition, the Bureau of Engineering, Navy Department, Washington, D. C., upon application, authorizes tests of the products of prospective bidders. After authorization, samples consisting of two bricks and two pounds of cement are forwarded to the Navy Yard, Philadelphia, where laboratory tests consisting of chemical analyses and fusion point determinations are conducted. If laboratory tests show conformity with chemical and physical requirements of specifications, additional samples consisting of fifty bricks and fifty

pounds of cement are submitted on which the simulative service test is conducted. By preceding the service test, which requires approximately one week for completion, by the more simple laboratory tests, the greatly inferior brands of brick and cement are generally eliminated.

The methods used, although not in themselves a solution of the refractories problems encountered, have proven invaluable for the purpose for which established, as the military necessities of the Navy cannot tolerate the use of furnace linings liable to failure at a time critical for the ship's safety.

NOTES.

A NOTE ON CRITICAL SPEEDS.

In view of the interest that has lately been shown in the question of torsional vibration on motor ships, the following information is noteworthy. It was found that if the propeller shaft were designed according to the rules of German Lloyds, the critical speed would be 75 R.P.M. It was thought that at full speed, *i.e.*, 95 R.P.M., and particularly at half-speed, this would give

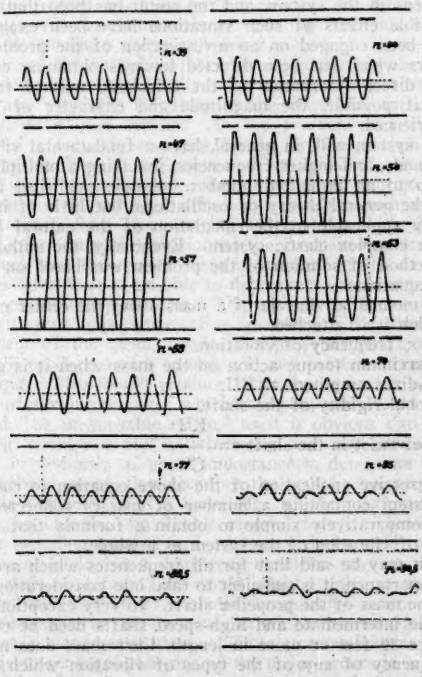


FIG. 7.—DIAGRAMS TAKEN WITH A FRAHM TORSIOGRAPH, SHOWING CRITICAL SPEEDS OF ABOUT 60 R.P.M.

rise to considerable vibration, so that it was arranged with German Lloyds to have a smaller propeller-shaft diameter, namely, 320 mm., instead of 360 mm. This gave a critical speed of 60 R.P.M., so that at full speed, *i.e.*, 95 R.P.M., half-speed (75 R.P.M.) and slow speed (45 R.P.M.) there would be practically complete absence of vibration. This is indicated in the torsional diagram illustrated in Fig. 7 taken with the Frahm indicator on the tunnel shaft.—"The Motorship."

TORSIONAL OSCILLATIONS AND MARINE REDUCTION GEARING.

By A. T. THORNE, B. Sc., and J. CALDERWOOD, M. Sc.

The importance of torsional vibrations has recently been emphasized by the fact that trouble experienced in connection with reduction gearing on board ship has, in one or two cases, been attributed to the existence of torsional vibrations in the system, and the result has been that in some quarters the possible effects of such vibrations have been exaggerated. The authors have been engaged on an investigation of the problem during the past two years which has been directed towards obtaining confirmation or otherwise by direct observation of the calculated natural frequencies and determining, if possible, the magnitude and character of the vibrations actually experienced.

Any elastic system will, in general, have a fundamental vibration involving only one node, and higher frequencies, involving a multiplicity of nodes, will occur up to an unlimited number. Ample published information is available on the general theory of oscillations, but little of it is in a form that is readily applicable to the calculation of the natural frequencies of vibration of a complex elastic system. Eventually, the authors found that the easiest method of solution of the problem was based on the following fundamental equation:—

I = the moment of inertia of a mass about the center of the shaft to which it is attached.

$K = 2\pi \times$ frequency of vibration.

Then the maximum torque action on the mass when it is given a vibration of two radians amplitude $= K^2I$.

C = torsional rigidity of the shaft.

Then angle of twist in the shaft $= \dfrac{K^2I}{C}$.

By the progressive application of the above equation to consecutive sections of a system containing a number of masses connected by flexible shafts, it is comparatively simple to obtain a formula that will give the natural period of vibration of the system as a whole.

In general, it may be said that for all frequencies which are likely to be of practical importance, it is sufficient to take into consideration the moment of inertia of the mass of the propeller shaft. In very exceptional cases only, the mass of the intermediate and high-speed shafts need be considered, as, unless these are 30 feet or more in length, their mass does not materially affect the frequency of any of the types of vibration which are likely to coincide with forced vibrations in the turbine, gearing or propeller.

There has been some criticism of the assumption that a system including gearing may be regarded as an unbroken elastic system. That it is a broken

system when at rest is obvious, but when transmitting a torque sufficient to ensure that the gear teeth are continuously in contact, such tooth contact is undoubtedly equivalent to a rigid connection.

If an elastic system is subject to the action of any periodic disturbing force, forced vibrations will be set up of a frequency equal to the frequency of the disturbing force. In the system we are considering, such disturbing forces may be due to external causes, of which the most prominent is the variation in propeller resistance, or may be due to what may be called "internal causes," such as slight lack of balance in any of the rotating masses of the system. When a forced vibration is impressed on the system, the character of the vibration set up will depend upon the point in the system at which the disturbing force is applied. Nodes will occur in positions such that the natural frequencies of the free ends of the system about those nodes—each free end being regarded as a fixed free system fixed at the node—are equal to the frequency of the impressed forced vibration. If the disturbing force is due to either of the causes mentioned above, this frequency will be directly proportional to the revolutions per minute of the propeller shaft, and consequently as the revolutions per minute rise the frequency of the forced vibration will increase. As this frequency increases, the nodes will move outwards towards the free ends of the system, and additional intermediate nodes will occur.

The amplitude of the forced vibration produced by any disturbing force in a given system depends primarily on the magnitude of the disturbing force and the ratio which the frequency of this disturbing force bears to the natural frequencies of the system. As the frequency of the disturbing force approaches a natural frequency of the system, the phenomenon of resonance occurs, resulting in an increase of the amplitude of the vibration. When the two frequencies coincide, the amplitude of vibration would become infinite but for the damping influences operating, such as the internal friction of the shaft, &c. The damping force varies with the amplitude, so that instead of an infinite vibration at the resonance point, we only get an amplitude such that the applied force is balanced by the damping forces. It is of obvious advantage to be able to determine a resonance factor without knowledge of the damping term. The multiplying factor due to resonance is of the utmost importance. For instance, if the disturbing torque is $\pm$ 10 per cent of the mean torque in a propeller shaft, a multiplying factor of 10 will produce a zero torque, and any further increase will mean a negative torque. This would produce tooth separation at each vibration, and the range of revolutions over which this happened would become what has been called "an unworkable range," as it is obvious that a system including gearing could not run satisfactorily for any period under such conditions. It is obviously of great importance to determine the resonance factor at the critical speed, thus enabling the actual resonance curve to be obtained, which, combined with a knowledge of the magnitude of the disturbing torque, enables the unworkable ranges, if any, to be determined.

In a system including gearing it is necessary that consideration be given to the positions of nodes, as a dangerous condition may occur if the node falls in either train of gears. This would imply a uniform angular velocity of these gears, and any slight periodic irregularity in the teeth—whose frequency corresponded to the frequency at which the node fell in the gears—would produce a very heavy stress. These stresses become much greater if the system is so arranged that one of the nodes occurs in a gear wheel at a natural vibration frequency.

Opportunities have now occurred for making actual observations on five single-screw vessels, and three twin-screw vessels, all fitted with geared

turbine installations; but details are given only of two twin-screw and two single-screw vessels. In all cases an endeavor has been made to obtain by direct reading the variation in torque with different positions of the propeller blade. The first step taken was the adoption of Messrs. Parsons' proposal to mount the torsionmeters, which were of the Hopkinson-Thring type, on a support, enabling them to be moved circumferentially round the shaft, the amount of movement being sufficient to obtain readings in all positions of a propeller blade, even with a three-bladed propeller. Owing to the construction of the torsionmeter, two sets of readings are obtained by this method, the interval between the two being approximately 150 degrees.

The taking of a set of readings from the torsionmeter occupied a considerable length of time, and it was found impossible to obtain during a trial trip curves at any considerable number of speeds. To overcome this difficulty, a vibration meter was designed which drew a curve of the vibration of the shaft over a period of one revolution. The instrument is similar to those which, we understand, are used by other firms. A comparatively heavy fly-wheel is mounted on ball bearings on a short length of shaft, and a very light aluminium pulley is similarly mounted on the same shaft. The pulley drives the fly-wheel through a light spring, and is itself driven by a belt off the main shafting. The pulley is so light that it follows exactly the movements of the main shaft, while the inertia of the fly-wheel is such that the slight variations of impulse transmitted through the spring are insufficient to disturb its uniform angular velocity. Vibrations of the shaft are thus reproduced by a relative movement of the pulley and fly-wheel. This relative movement is transmitted through a lever carrying a pencil which draws a circular diagram. The pencil lever was arranged to give various magnifications to the vibration so that a wide range of amplitudes of vibration could be measured. It was found with the big magnification required for small vibrations that pencil friction was so great as to interfere seriously with the accuracy of the readings taken, and an electrical method of recording, designed by Mr. J. C. Arthur, was used. The curve of vibration in this method was drawn by the perforation of the diagram paper by a series of sparks passing from a sharp point on the pencil arm to an insulated plate behind the paper.

The diagrams drawn by the vibration meter show the vibrations at one position of the shaft, from which a curve of torque variation can be drawn by dividing the curve obtained into its constituent vibrations and calculating the torque variation of each of these components from the position of its node.

Case A.—This was a twin-screw vessel, 450 feet p.p. in length, with a block coefficient of about 0.66. The machinery consisted of double-reduction geared turbines developing about 8,000 shaft horsepower at 130 revolutions per minute. The speed of the vessel under these conditions was about 17 knots. The propellers were of the built type, each having four manganese bronze blades.

In this case, the natural frequency of the system corresponded to the vibration arising at a propeller speed of 45 revolutions per minute, with a four per revolution vibration. The amplitude of vibration was so small that accurate curves could not be obtained from torsionmeter readings. Curves were, however, obtained by the aid of the vibration meter, but while the amplitude of the four per revolution frequency could be obtained from these curves, it was impossible to separate any of the other components of the vibration. Fig. 1 is a curve taken at 110 revolutions per minute corrected for the resonance factor of the four per revolution vibration; an amplitude varying between 8 and 12 per cent is shown by this curve. The unequal heights of the four waves indicate the presence of another vibra-

tion of slightly different frequency, probably arising in the gearing. Fig. 2 is a curve taken at 69 revolutions per minute corrected for resonance, and shows the same characteristics and almost the same amplitude.

Case B.—This was a twin-screw vessel of 465 feet p.p., with a block coefficient of about 0.73. The machinery consisted of double reduction geared turbines and a power of about 4,300 shaft horsepower was developed at 86 revolutions per minute, the speed of the vessel under these conditions being about 13½ knots. The propellers were of the built type, each having three manganese-bronze blades. In this case, torsionmeter readings were unfortunately only obtained over a comparatively small angle, and as the vibration meter had not then been constructed, no very complete records were obtainable. Fig. 3 gives the curve corrected for resonance, and indicates a three per revolution vibration of an amplitude of 9 per cent with a slight indication of a six per revolution vibration whose amplitude is too small for definite measurement.

Case C.—This was a single-screw oil tanker with machinery aft, the length being 440 feet p.p. The machinery consisted of double reduction geared turbines and a three-bladed propeller of the built type was fitted. The power developed was about 3,100 shaft horsepower at about 75 revolutions per minute, the speed being about 11¼ knots.

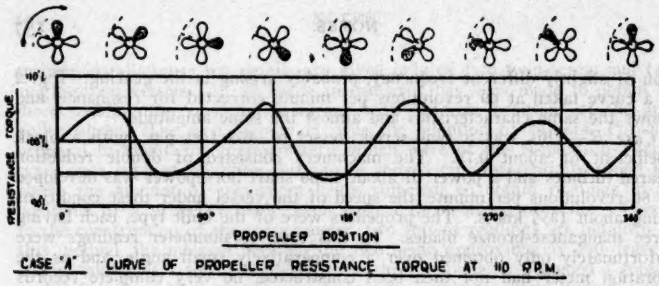
In the case of this vessel, the first natural frequency of the system corresponded to revolutions of only some 10 per cent below full-power revolutions with a four-bladed propeller. Under these circumstances the shaft diameter was increased and a three-bladed propeller fitted to remove the natural period further from the running revolutions.

Curves were plotted from readings taken with the tips of the propeller blades just immersed. The torsionmeter scale box was moved 5.8 degrees after each reading, a total angle of 94.5 degrees being possible. The readings from one mirror of the torsionmeter were separated from those of the other mirror by 153 degrees. The corrected torsionmeter readings were plotted on a base of propeller angle, a blade being vertical on top at 0, 120, 240 and 360 degrees.

The torsional vibrations caused by periodic torque variations will be due to the change in resistance caused by the passage of a blade through the upper portion of the aperture, and to a greater or less extent by the passage of a blade through the lower half of the aperture, considerable difference of opinion having been expressed as to the relative importance of the latter variation. If the former only is of importance, this will result in the principal vibration having a frequency of three times the revolutions; if both impulses—due to the top and bottom passage—are approximately equal, the resulting vibration will have a frequency six times the revolutions. These correspond to torque variations with periods of 120 degrees and 60 degrees of propeller angle, respectively.

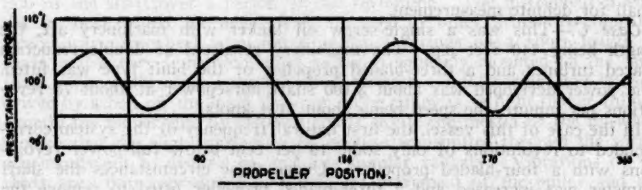
In the curve shown in Fig. 4 the points obtained from the mean values of readings are joined by a curve drawn in full line, and the shape of the curve between 167 degrees and 280 degrees and between 14 degrees and 73 degrees is assumed and shown by a dotted line.

The resulting curve is evidently composed of a 60 degree period vibration superposed on another vibration with a period of 120 degrees. The two vibrations are assumed to follow approximately a sine law, and on this assumption the 120 degrees period vibration is shown in chain dotted line. The amplitude of the two vibrations can now be obtained, and dividing these amplitudes by their resonance factors, values of the torque variation at the propeller are obtained, these being 9.5 per cent for the 120 degree period vibration and 11 per cent for the 60 degree period vibration.



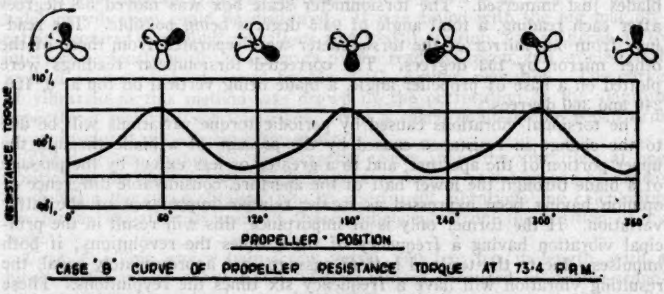
CASE 'A' CURVE OF PROPELLER RESISTANCE TORQUE AT 110 R.P.M.

FIG 1



CASE 'A' CURVE OF PROPELLER RESISTANCE TORQUE AT 69 R.P.M.

FIG 2.



CASE 'B' CURVE OF PROPELLER RESISTANCE TORQUE AT 73·4 R.P.M.

CASE 'C' PROPELLER TORQUE

FIG. 3.

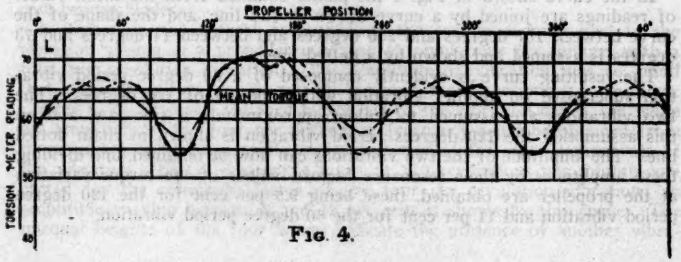


FIG. 4.

CASE 'D' TORQUE VARIATION ON MOORINGS TRIAL.

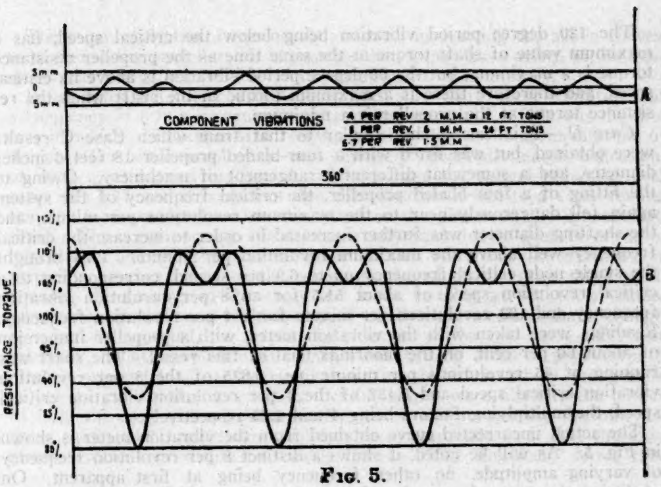
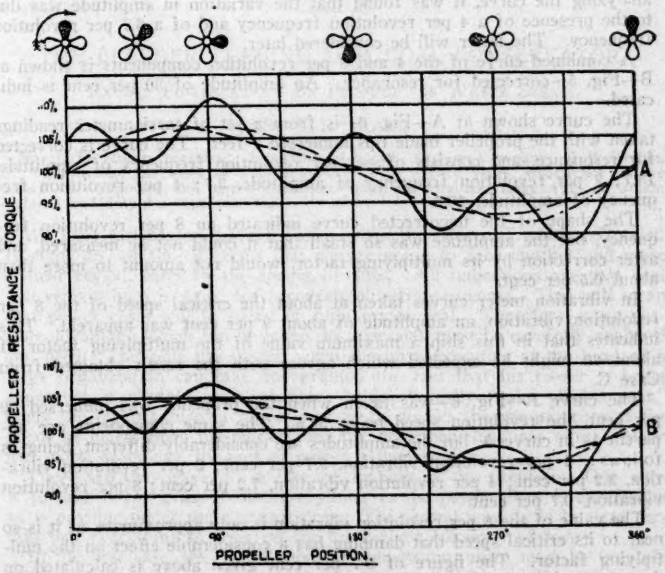


Fig. 5.



CASE 'D' CURVE 'A'- PROPELLER RESISTANCE TORQUE AT 73·5 R P M
" 'B'- " " " " 54·5 "

Fig. 6

The 120 degree period vibration being below the critical speed, has a maximum value of shaft torque at the same time as the propeller resistance torque is a maximum, but the 60 degree period vibration is above its critical speed, and therefore there is a maximum torque in the shaft when the resistance torque at the propeller is a minimum.

Case D.—This was a ship similar to that from which Case C results were obtained, but was fitted with a four-bladed propeller 18 feet 6 inches diameter, and a somewhat different arrangement of machinery. Owing to the fitting of a four-bladed propeller, the critical frequency of the system again fell dangerously near to the maximum revolutions per minute, and the shafting diameter was further increased in order to increase the critical frequency well above the maximum revolution per minute. This brought the single node critical frequency up to 6.9 per second, corresponding to a critical revolution speed of about 51.5 for an 8 per revolution vibration frequency and 103 revolutions per minute for a 4 per revolution frequency. Readings were taken with the vibration meter, with a propeller immersion of about 60 per cent, on the moorings trial of this vessel. The shaft was running at 45 revolutions per minute, *i.e.*, 0.875 of the 8 per revolution vibration critical speed and 0.437 of the 4 per revolution vibration critical speed, the multiplying factors being 4 and 1.24 respectively.

The actual uncorrected curve obtained from the vibration meter is shown in Fig. 5. As will be noted, it shows a distinct 8 per revolution frequency of varying amplitude, no other frequency being at first apparent. On analyzing the curve, it was found that the variation in amplitude was due to the presence of a 4 per revolution frequency and of a 6.7 per revolution frequency. The latter will be considered later.

A combined curve of the 4 and 8 per revolution components is shown at B—Fig. 5—corrected for resonance. An amplitude of 30 per cent is indicated.

The curve shown at A—Fig. 6—is from a set of torsionmeter readings taken with the propeller blade tips immersed 7 feet. The curve is corrected for resonance, and consists of:—1 per revolution frequency of amplitude, 13.7; 2 per revolution frequency of amplitude, 3.7; 4 per revolution frequency of amplitude, 10.5.

The shape of the uncorrected curve indicated an 8 per revolution frequency, but the amplitude was so small that it could not be measured, and after correction by its multiplying factor, would not amount to more than about 0.5 per cent.

In vibration meter curves taken at about the critical speed of the 8 per revolution vibration, an amplitude of about 9 per cent was apparent. This indicates that in this ship a maximum value of the multiplying factor of about 20 might be expected which agrees with the result obtained from Case C.

The curve B—Fig. 6—was taken when the propeller was immersed 90 per cent, the revolution speed being 54.5. The same components are apparent as in curve A, but the amplitudes are considerably different, being as follows:—1 per revolution vibration, 9.7 per cent; 2 per revolution vibration, 3.2 per cent; 4 per revolution vibration, 7.2 per cent; 8 per revolution vibration, 0.7 per cent.

The value of the 8 per revolution vibration is only approximate, as it is so near to its critical speed that damping has a considerable effect on the multiplying factor. The figure of 0.7 per cent given above is calculated on the assumption of a maximum multiplying factor of 20 at the resonance point, this giving a factor of 6.5 at the speed of 54.5 revolutions per minute. It must again be remembered that the possibility of so large a multiplying factor as 20 is due to the altogether abnormal circumstances existing in this vessel.

The 1 per revolution frequency is assumed, in drawing the above curves, to be due to the propeller. It is possible that it may have arisen in the low-speed gear wheel, but in view of its magnitude this is unlikely.

In all vibration meter diagrams on this ship there occur vibrations of a frequency of 6.72 per revolution of the propeller shaft. This was the first case in which a vibration of a frequency other than an integral multiple of the propeller shaft revolutions was definitely observed, though indications of such were noticed in Case A. In this case the second reduetion gear ratio was 6.72, and it is evident that the vibration is due to a 1 per revolution tooth inequality in either the wheel or the pinion on the intermediate shaft. The occurrence of vibrations directly traceable to gear errors is of importance, and the observations made on this vessel have been fully confirmed in another vessel of the same type, in which diagrams taken while running the gearing in the shop and while running at sea show similar vibrations, synchronous with the speed of the intermediate shaft, though, as might be expected, the amplitudes observed in the two cases differ considerably. The amplitude of a vibration arising from a gear error is directly proportional to the rigidity of the shafts on which the gear is mounted. In the case in question, flexible shafts were fitted to the pinions of both the first and second reduction gears, and it was the flexibility of these shafts that limited the amplitude to a magnitude such that the tooth stresses were kept within reasonable limits.—"The Engineer," Nov., 1922.

THE BLEEDER TURBINE AS A COAL CONSERVATOR.

Excess Power as a "By-Product" of Waste Steam.

By J. L. Moore.

Mr. J. L. Moore, president of the Moore Steam Turbine Company, of Wellsville, N. Y., preceded his work as a manufacturer with twenty years' experience in the design of compound and triple expansion pumping engines, large Parsons steam turbines and Impulse type turbines.

We have just passed through a coal crisis, the effects of which still exist. If signs do not fail, it seems that we are going to face another and more critical period, early in the spring of 1923. All indications are that coal will remain high in price for a considerable time to come. This will mean that power costs will be based on high priced coal. Therefore every possible advantage which can be taken, to save coal, must be carefully analyzed and used. The time has passed when the busy executive at the head of a large organization can take for granted the fact that his power is being generated as cheaply as is possible; he must really "take off his coat," study the manufacture of power just as he studies the manufacture of the goods he places upon the market, and put his conclusions into effect. More and more, every day, the cost of power is increasing.

There are two ways, of course, of looking at the situation. One way, is to take it for granted that everything in the power plant is as efficient as is possible under the circumstances. The second way, is to be dead certain that everything is designed for the highest efficiency of *today*.

In the first case, the installation of generating units may be very old style, and yet as far as they alone are concerned they may be operating at their highest point of efficiency. On the other hand, regarding it from the standpoint of the second case, the old existing installation may have been all right ten years ago; but today it may be all wrong.

A great many of our industries today would be considerably ahead if they would "clean house" in their power plants and replace old machinery, which was satisfactory ten years ago, with machinery which is designed for present day requirements.

One of the most recent advances which has been made in steam turbine design in the power plant, is the bleeder or extraction type of turbine. It is not a new idea, because back as far as 1889 the idea itself is thoroughly described by Mr. Charles T. Main in Vol. 10 of the transactions of the A. S. M. E. In this case, Mr. Main applied the idea to steam engines. The operation, however, lends itself so admirably to steam turbine design that it has found immediate favor among that class of industries that can use it.

The bleeder turbine is so named because it "bleeds off" steam which has already done work, for other purposes rather than power; which may be process work, heating, or cooking. Therefore the "bleeder" turbine is adapted particularly to those industries which have use for steam for heating, cooking, or process work. Industries of that nature, which utilize the "bleeder" idea, will find that they will save an enormous amount of steam which at the present time is going into the air. Putting it another way, they will reduce the consumption of coal so very materially that results may prove to be startling. It is not an unheard of thing for an installation of a bleeder turbine, in a plant in which it is applicable to show at least 100 per cent investment return upon the money expended for it.

The bleeder turbine consists of a high pressure, non-condensing turbine, expanding the steam from the initial pressure to somewhere near atmospheric pressure (usually 5 pounds to 30 pounds gauge) built on the same shaft with, and exhausting into, a low pressure turbine which expands all or part of the steam from the receiver or bleeder pressure to the vacuum in the condenser. A blank diaphragm (see Fig. 1) separates the non-con-

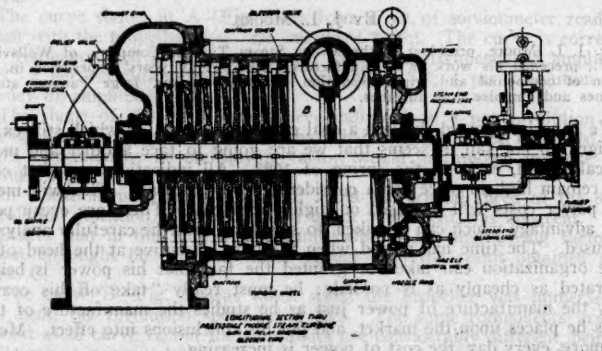


Fig. 1. Cross-section of a bleeder turbine, showing the diaphragm which separates the non-condensing high-pressure turbine (A) from the low pressure turbine (B)

densing high pressure turbine (A) from the low pressure turbine (B). Between the high pressure non-condensing turbine and the low pressure turbine the bleeder connection is made and the steam is extracted for heating, cooking or other process work. The turbine, of course, drives a generator.

The bleeder pressure is maintained constant, or nearly so, by an automatic cylindrical valve (see Figs. 1 and 2), actuated by any slight change of bleeder pressure. The automatic valve referred to controls steam from the receiver (A) to the low pressure turbine (B). In other words, steam at boiler pressure passes through the first high pressure stage of the turbine and does work, and is expanded down to the desired bleeder pressure.

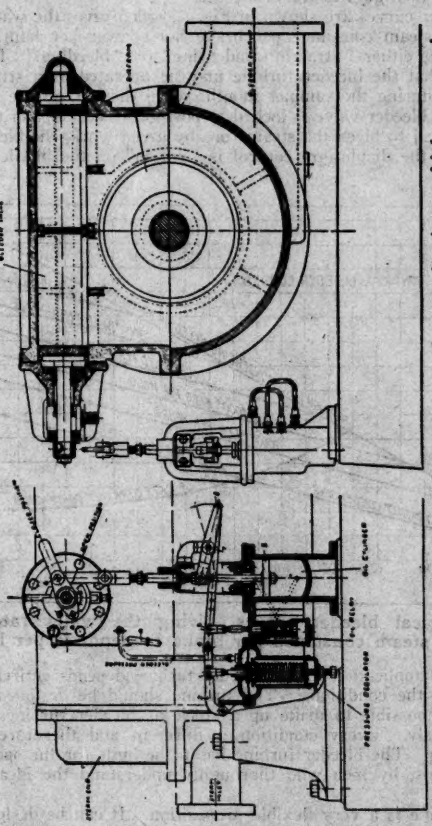


Fig. 2. Detailed cross-section of the bleeder valve mechanism, which automatically controls steam from the receiver (A) to low-pressure turbine (B).

This turbine, therefore, is nothing more or less than a power generating, reducing valve. It takes the place of the present style reducing valve which is used in so many industries in this country. It reduces pressure from boiler pressure down to the desired back pressure, but in so doing, furnishes power with which the factory is operated. The old style reducing valve throws this power away—blows it to the air.

Bleeder turbines may also be designed for extracting steam at two or more different pressures, if desired. For instance, in plants where steam for a certain process is required, at say 60 pounds gauge pressure, and for another process or heating the building, at 5 pounds pressure, two automatic bleeder valves will be required, to maintain the desired pressures. The excess steam then passes through the remaining stages of the turbine, doing more work, and then goes to the condenser.

Typical bleeder curves are shown in Fig. 3, which gives the water rate or the amount of steam consumed per brake horsepower per hour, with the turbine operating either "straight condensing" or "bleeding." It is interesting to note that the bleeder turbine may be operated as a straight condensing turbine during the summer months when there is no use for steam. In this case, the bleeder valve is locked in position to accomplish this result. When it is desired to bleed the steam for any given work, the bleeder valve is unlocked and the diaphragm control is "cut in" on the bleeder pressure system.

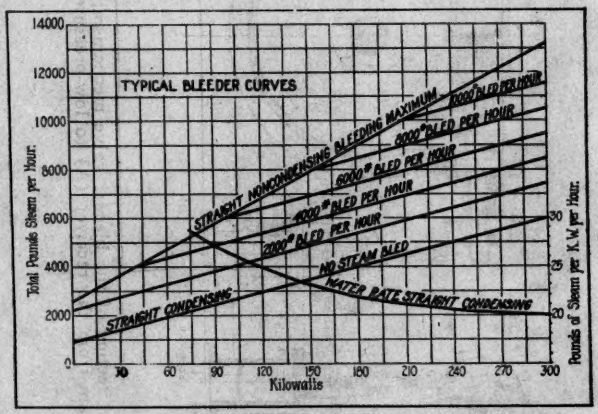


Fig. 3. Typical bleeder curves, giving the water rate or amount of steam consumed per brake horsepower per hour.

The successful application of the bleeder turbine depends entirely upon a careful study of the conditions. The turbine should be *designed* to meet them. It is not possible to make up a stock of bleeder turbines and sell them promiscuously. Every condition is different and therefore receives different handling. The bleeder turbine has to be built for the specific, existing requirements, by men who thoroughly understand the idea and the design back of it.

A bleeder turbine is a very flexible proposition. It can be designed as a straight high pressure condensing turbine, and at the same time any amount of steam within reasonable limit can be bled by the use of extra nozzles on the first stage under control of hand valves. Or, if it is desired, a group of extra nozzles under the control of a single automatic overload valve may be furnished. Hand valve control of the extra nozzles is satisfactory if the steam bled is practically constant. If it varies considerably and at frequent intervals, or irregular intervals, then an automatic overload valve is

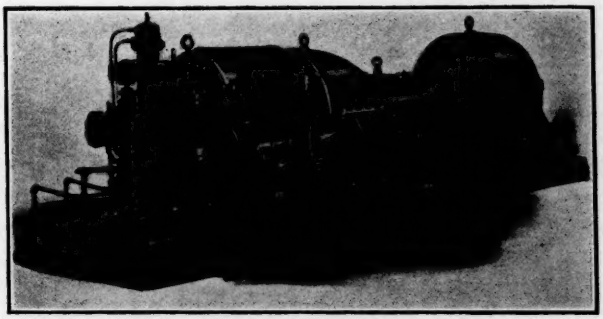


FIG. 4.—BLEEDER TURBINE, SHOWING AUTOMATIC OVERLOAD VALVE IN THE FOREGROUND.

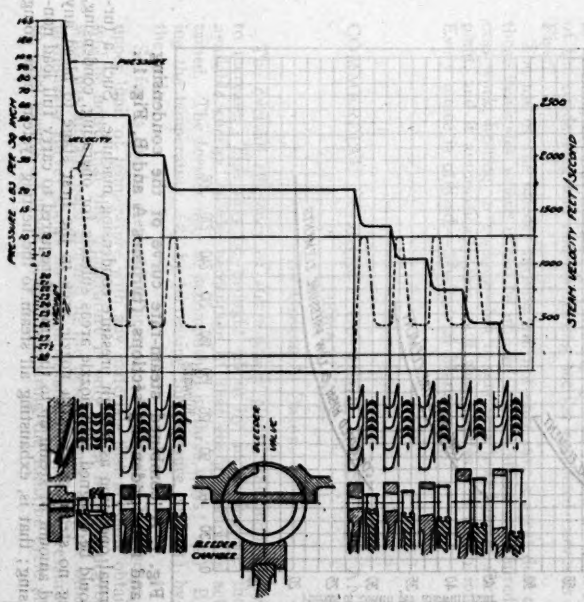


Fig. 5. Bleeder valve placed after the second Rateau stage. The velocity wheel, with two rows of buckets, is clearly shown here.

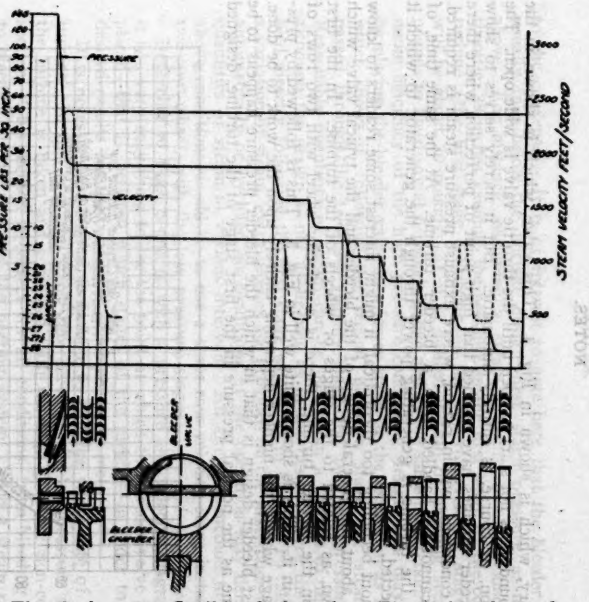


Fig. 6. A more flexible design than Fig. 5 is where the steam is bled directly following the first stage wheels, as shown.

necessary, which is shown in Fig. 4. This valve, which is shown in the foreground, does not start to open until the throttle valve is wide open. The operation, of course, is entirely automatic. But it merely serves to show that bleeder turbines have reached that high point of perfection where there are no conditions, within reason, where exhaust pressure steam is required, which cannot be handled with a bleeder turbine. At the same time, of course, the turbine is generating power through the generator to which it is connected.

Without becoming too theoretical, it might interest some readers to know a little about the general design of the bleeder and the typical valve which is shown, as related to the stages or wheels of the turbine. In the first place, in the bleeder turbine, there is a velocity wheel with two rows of buckets in it. This is shown quite well in Fig. 5. This is followed by pressure stage wheels, the number depending entirely upon the work to be done. The best bleeder design is that in which the bleeder pressure happens to be the same as the normal pressure in the first stage of the turbine, designed

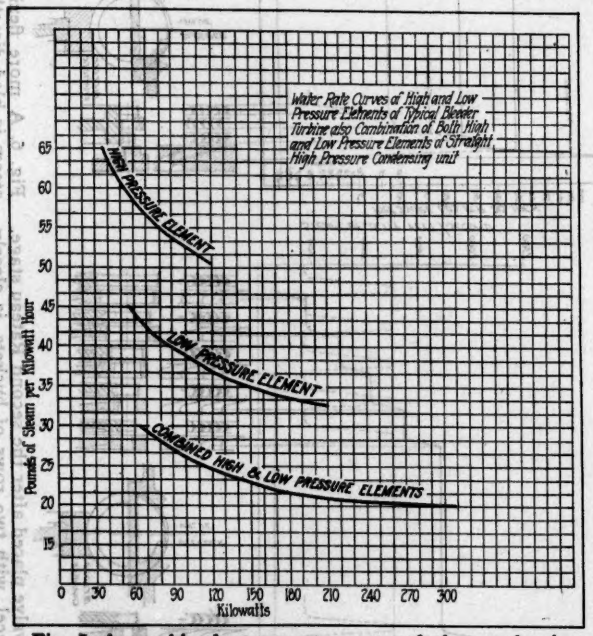


Fig. 7. A combined steam-rate curve of the condensing and non-condensing sections, (turbines A and B, Fig. 1).

for normal operation as a high pressure condensing machine. Such a turbine would be designed with nozzle areas suitable for operating, condensing, bleeding no steam, with extra nozzle area in the first stage to bleed any required amount of steam up to the amount required to carry full load non-condensing; that is, exhausting all steam to the heating system, none going

to the condenser. It is of interest to note here that the bleeder pressure usually falls below the normal design first stage pressure, as mentioned above. For instance, assuming a steam pressure of 150 pounds at the throttle, 28 inches vacuum, the pressure in the first stage of the condensing turbine (the first stage consisting of the two-row velocity stage wheel) will be about 30 pounds. If it is desired instead of bleeding steam at 30 pounds to bleed say at 5 pounds gauge and at the same time retain the maximum efficiency of the high pressure condensing design, this may be done by placing the bleeder valve in a lower stage of the expansion as in Fig. 5, which shows it following the second pressure stage, the steam having passed through the two-row velocity stage and two low pressure stages. Reasonable quantities of steam can be bled from a turbine of this design although the design is not as flexible as in the design where the steam is bled directly following the first stage wheels, as in Fig. 6. To take care of a variation of steam, extra nozzles are provided on the first stage, controlled automatically or by hand, as conditions of the operation may demand, for quantities of steam at various intervals. An alternate method to meet the above conditions would be to use a three-row velocity stage instead of the two-row wheel, with the low pressure stages.

Referring to Figs. 5 and 6, with a pressure and velocity diagram given, it is a very easy thing to see the point at which the bleeder valve should be placed—that is, between which stages of the turbine—in order to give the desired results for the conditions which are to be met.

It is necessary, of course, to be able to plot the steam rate of a bleeder turbine. In this case, the water rate curve of the low pressure section of the turbine is taken and combined with the high pressure non-condensing section. This method is correct for all conditions, from bleeding no steam to bleeding all the steam to the heating system. Such a steam rate curve of the condensing and non-condensing section combined curves is shown in Fig. 7.

The use of the bleeder turbine will not only save a great deal of coal in those industries where it can be utilized—where steam is required for process work, heating, or cooking—but it acts as a heat balance for the entire plant, and it automatically sees to it that no steam is blown into the air. Every ounce is utilized.—"Industrial Management."

COMMERCIAL ECONOMY OF HIGH PRESSURES AND TEMPERATURES.

By George A. Orrok.

The over-all efficiency of steam-power plants will undoubtedly continue to increase. This may be brought about by increasing the theoretically available energy, or the percentage of that energy which is actually obtained. The former may be accomplished in three ways: (1) By increasing the temperature, (2) by increasing the pressure, and (3) by changing the cycle.

So far as can be seen at the present, 1,100 degrees F. appears to be the upper limit of temperature with any materials that are now obtainable. It will be noted that this temperature is only a few hundred degrees beyond what has already been attained in the newer plants. On the other hand, a preliminary survey shows no insurmountable obstacles to pressures of 1,200 to 1,500 pounds. Curves plotted, comparing the Rankine cycle and the re-

generative cycle, show a distinct thermal advantage for the latter at all temperatures and pressures. This advantage increases only slightly with increase of temperatures at constant pressure, but considerably with increase of pressure at constant temperature.

After allowing for the estimated increase in investment charges, preliminary calculations seem to show that so long as the Rankine cycle is used, there can be little, if any, economic gain by increasing either pressure or temperature beyond present commercial limits. This conclusion is, of course, subject to modification in cases of changes in the relative cost of fuel and equipment.

The regenerative cycle, on the other hand, seems to show a distinct economic advantage over the Rankine, even at 700 degrees. This advantage increases with increased pressure and is still increasing rapidly at 1,200 pounds. There seems to be a slight additional net gain at all pressures by using 1,000 degrees temperature instead of 700, but the difference is not great on paper and might easily prove to be non-existent in practice.

The conclusion is drawn that, so long as we stick to the present cycle, our chief hope of increased economy must lie in perfecting details and obtaining a larger percentage of the theoretically available heat. On the other hand, it seems quite possible that there may be an economic future for pressures as high as 1,200 pounds in plants operating on the regenerative cycle.—"Power."

FEED HEATING FOR HIGH THERMAL EFFICIENCY.

By Linn Helander.

This paper considers, first, for a theoretically perfect plant and secondly for an actual plant of 25,000 kw., the effect of feed temperature upon plant efficiency, the object being to determine the best feed temperature from the thermal point of view. No account is taken of investment or operating costs.

Various methods of feed-water heating are considered, including single-stage heating by house-turbine exhaust and single- or multiple-stage heating by steam bled from the main unit.

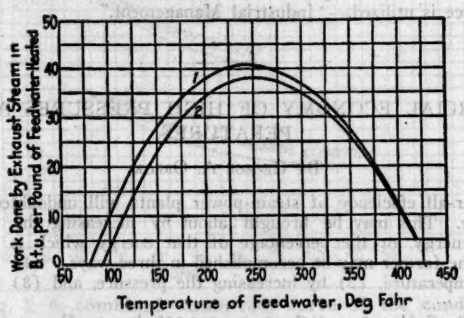


Fig. 1.—Relation of Feed Temperature to Work Done by Heating Steam In Theoretical Plant.

Considering first a theoretically perfect plant with single-stage heating by the exhaust of a house turbine, it develops that the best plant thermal efficiency comes with a feed-water temperature such that the power ob-

tained from the house turbine is a maximum. This power is the product of the power generated per pound of steam supplied the house turbine by the number of pounds so used. The two limiting feed temperatures, that of the main condensate and that in the boiler, give zero power because, in the first case no heating steam is used, and in the second no expansion or pressure drop is possible.

The maximum house turbine power must come at an intermediate point about halfway between these two temperatures, as shown by the curves in Fig. 1. These give the theoretical work obtained from the heating steam with various feed temperatures and with the initial steam at 325 pounds gauge pressure and 200 degrees superheat. Curve 1 is for a 29-inch vacuum (main condensate at 79 degrees F.), while curve 2 is for a 28½-inch vacuum (92-degree condensate). The first case shows the best feed-water temperature around 240 degrees, and the second around 260 dgrees.

A similar curve drawn for two-stage heating (as by two bleeder heaters) with equal heating per stage, shows a best feed-water temperature of 300 degrees for the 29-inch vacuum. The more stages used the higher is this best feed-water temperature. With an infinite number of stages it becomes that of the boiler, and the plant efficiency equals that of the Carnot cycle. It must be remembered that this applies to a theoretically perfect plant. An actual plant gives the best thermal economy at somewhat lower temperatures.

Fig. 2 gives the predicted actual plant heat consumption per kilowatt-hour

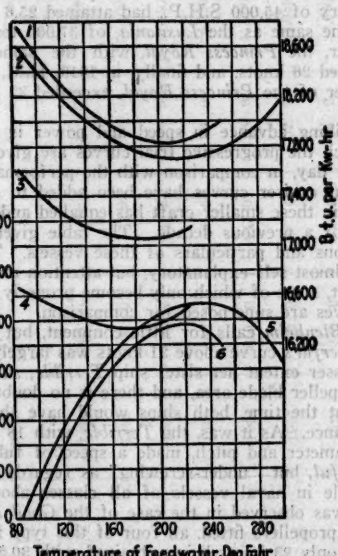


FIG. 2.—EFFECT OF FEED TEMPERATURE UPON PLANT EFFICIENCY AND LOAD CARRIED BY HEATING STEAM FOR ACTUAL 25,000-KW. PLANT.

for a 25,000-kw. plant under various conditions. Curve 1 is for a plant without economizers, using a 1,600-kw. house turbine and not bleeding the main unit, while curve 2 is for a plant without economizers, using a 650-kw. house turbine and supplementing its exhaust with steam bled from the main unit. Curve 3 is the same as curve 2, except that an economizer is used, while curve 4 is the same as curve 3, but with twice the amount of economizer surface. Curves 5 and 6 show the loads developed by the heating steam, the first being for the 650-kw. house turbine with bleeding, while the latter is for the 1,600-kw. house turbine without bleeding.—"Power."

SOME INTERESTING SPEED AND POWER CURVES.

The Striking Advance In Speed and Power Made During the Past Thirty Years.

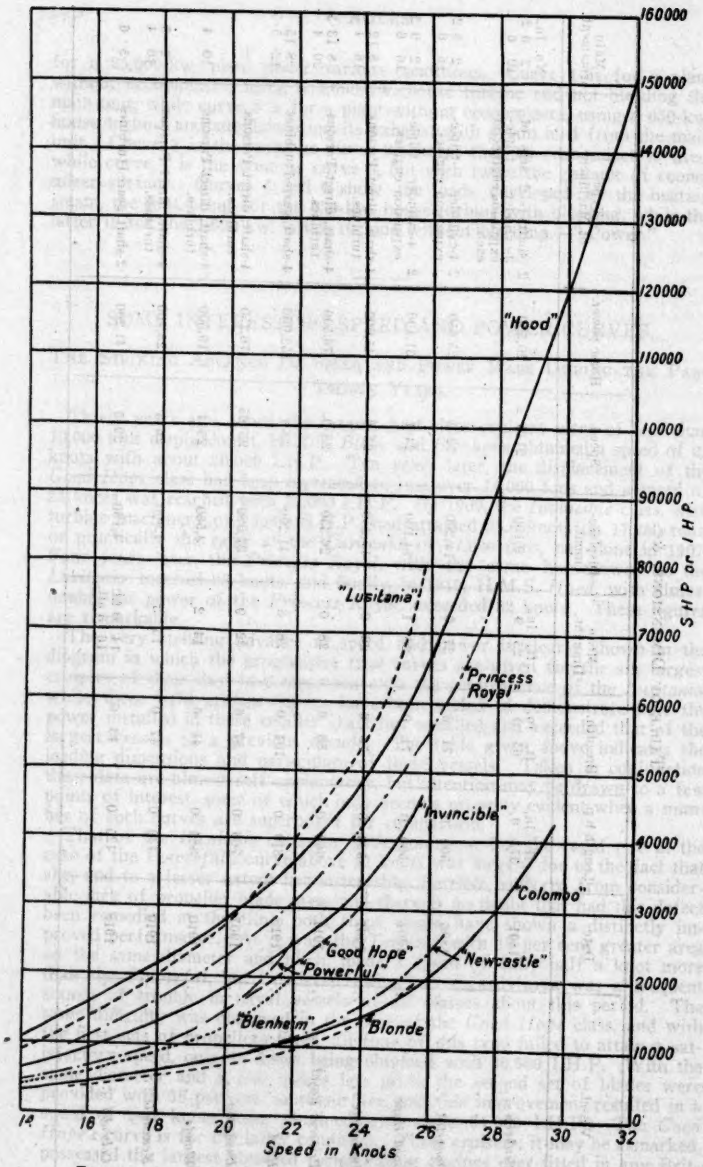
Thirty years ago, when the largest first-class cruisers were of less than 10,000 tons displacement, H.M.S. *Blake* and *Blenheim* obtained a speed of 22 knots with about 20,000 I.H.P. Ten years later, the displacement of the *Good Hope* class had been increased to just over 14,000 tons and a speed of 24 knots was reached with 31,000 I.H.P. By 1909, the *Invincible* class, with turbine machinery of 45,000 S.H.P., had attained 25.6 knots on 17,250 tons, or practically the same as the *Lusitania*, of 37,000 tons, had done in 1907. Four years later, the *Princess Royal*, with the same horsepower as the *Lusitania*, touched 28 knots, and finally, in 1919, H.M.S. *Hood*, with almost double the power of the *Princess Royal*, exceeded 32 knots. These figures are remarkable.

The very striking advance in speed and power is clearly shown in the diagram in which the progressive trial curves are given for the six largest cruisers of their day, in comparison with the performance of the *Lusitania*, while three light cruiser curves have been added to demonstrate how the power installed in these smaller craft has equalled and exceeded that of the largest vessels of a previous decade. The table given above indicates the leading dimensions and particulars of these vessels. Taken in conjunction these data are almost self-explanatory, but attention may be drawn to a few points of interest, some of which only become properly evident when a number of such curves are superposed for comparison.

That of the *Blenheim* calls for little comment, but the rapid rise in the case of the *Powerful's* curve above 21 knots was largely due to the fact that she, and to a lesser extent her sister ship *Terrible*, suffered from considerable lack of propeller blade area, and there is no doubt that had this defect been remedied at the time, both ships would have shown a distinctly improved performance. As it was, the *Terrible*, with 18 per cent greater area on the same diameter and pitch, made a speed of fully half a knot more than the *Powerful*, but "under-screwing" as regards area was a frequent source of trouble in naval vessels of all classes about this period. The same difficulty was observed in the case of the *Good Hope* class, and with the first sets of propellers fitted, all four of this type failed to attain a satisfactory speed, only 23 knots being obtained with 30,500 I.H.P. With the same diameter and a few inches less pitch, the second set of blades were provided with 38 per cent more surface, and this improvement resulted in a speed of 24.1 knots being obtained with about 31,000 I.H.P.. The *Good Hope's* curve is for the latter condition. These cruisers, it may be remarked, possessed the largest powered reciprocating engines ever fitted in any Brit-

Table of Dimensions, &c.

Vessel.	Date.	Length b.p.	Breadth.	Mean Draught.	Displacement.	Speed.	Horse power.	Type of Machinery.	Main Armament.
		ft. in.	ft. in.	ft. in.					No. In.
H.M.S. *Blenheim*	1891	375 0	65 0	25 9	9,100	21·0	20,500	2 shafts: 2 sets of 3-cylinder triple-expansion engines on each.	2 9·2 10 6
H.M.S. *Powerful*	1896	500 0	71 6	27 0	14,200	22·0	25,650	2 4-cylinder triple-expansion engines.	2 9·2 16 6
H.M.S. *Good Hope*	1901	500 0	71 0	26 1	14,100	24·1	31,400	2 4-cylinder triple-expansion engines.	2 9·2 16 6
H.M.S. *Invincible*	1907	530 0	78 6	26 0	17,250	25·6	45,000	4-shaft, direct Parsons turbines.	8 12 16 4
H.M.S. *Princess Royal*	1911	660 0	88 6	28 0	26,350	28·0	76,500	4-shaft, direct Parsons turbines.	8 13·5 20 4
H.M.S. *Hood*	1918	810 0	104 0	28 6	41,200	32·1	152,000	4-shaft, geared turbines	8 15 12 5·5
H.M.S. *Lusitania*	1907	760 0	87 6	32 9	37,080	25·65	76,250	4-shaft, direct Parsons turbines.	—
H.M.S. *Blonde*	1910	385 0	41 6	14 2	3,560	25·4	19,500	4-shaft, direct Parsons turbines.	10 4
H.M.S. *Newcastle*	1910	430 0	47 0	15 3	4,800	26·3	25,400	4-shaft, direct Parsons turbines.	2 6 10 4
H.M.S. *Colombo*	1918	425 0	43 6	14 0 fd. 16 3 aft.	4,190	29·5	41,200	2-shaft, geared turbines	5 6



Progressive Trial Curves of Various Types of Cruisers and of the "Lusitania."

ish warship. Actually, it was not the largest total power fitted in any cruiser driven with piston-type machinery, the French *Ernest Renan* having developed nearly 37,000 I.H.P. on trial, but the latter, as well as some of the last of the German cruisers, prior to the adoption of turbine machinery and which, like the *Blucher*, were heavily forced on trial, were triple-screw vessels, and the power per shaft was not as big as in the twin-screw *Good Hope*. The only British mercantile vessels which approached the horsepower of this class were the *Campania* and *Lucania*, and their power was not exceeded until some years later in the big German liners.

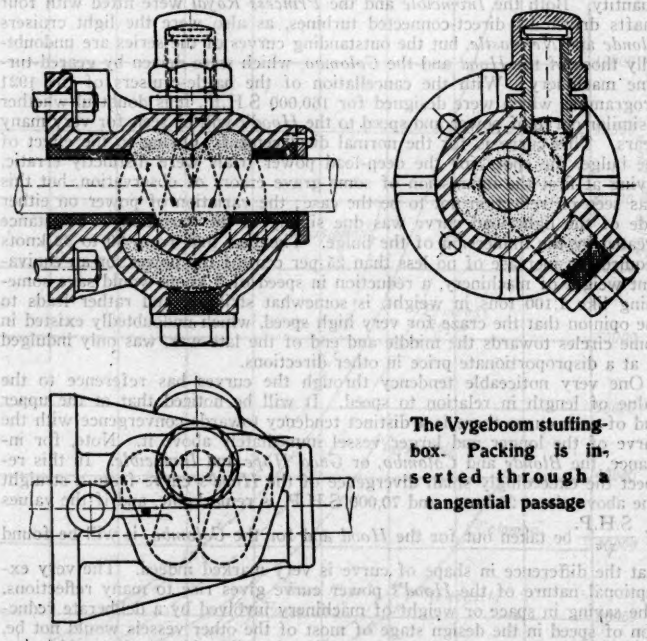
It is all the more interesting, therefore, to note that the next power curve in order of date was that of the *Lusitania*, very full details of which trials were published by Sir Thomas Bell in 1908. The *Invincible* class ran their trials in the following year, and it is probable that higher speeds might have been obtained easily had it not been for the policy then followed of restricting the horsepower developed on trial to about 5 per cent above the specified quantity. Both the *Invincible* and the *Princess Royal* were fitted with four shafts driven by direct-connected turbines, as also were the light cruisers *Blonde* and *Newcastle*, but the outstanding curves of the series are undoubtedly those of the *Hood* and the *Colombo*, which were driven by geared-turbine machinery. With the cancellation of the battle-cruisers of the 1921 programme, which were designed for 160,000 S.H.P., it is doubtful whether a similar curve of power and speed to the *Hood's* will be seen for very many years. This curve is for the normal displacement. Owing to the effect of the bulge, the spots for the deep-load power trials were distinctly erratic, giving at first the impression of some grave errors of observation, but this was seen on reflection not to be the case; the variation of power on either side of the light load curve was due simply to the variation of resistance created by the immersion of the bulge. The fact that from 30 to 32 knots requires an increase of no less than 25 per cent of the power, or an equivalent weight of machinery, a reduction in speed of 2 knots would save something like 1,100 tons in weight, is somewhat striking, and rather leads to the opinion that the craze for very high speed, which undoubtedly existed in some circles towards the middle and end of the late war, was only indulged in at a disproportionate price in other directions.

One very noticeable tendency through the curves has reference to the value of length in relation to speed. It will be noticed that at the upper end of each curve there is a distinct tendency towards convergence with the curve of the longer and larger vessel immediately above it. Note, for instance, the *Blonde* and *Colombo*, or *Good Hope* and *Invincible*. In this respect the exceedingly small divergence of the *Hood's* curve from a straight line above about 26 knots and 70,000 S.H.P. is remarkable, and if the values of $\frac{\text{S.H.P.}}{V^3}$ be taken out for the *Hood* and for the *Colombo*, it will be found that the difference in shape of curve is very marked indeed. The very exceptional nature of the *Hood's* power curve gives rise to many reflections. The saving in space or weight of machinery involved by a deliberate reduction of speed in the design stage of most of the other vessels would not be, on the whole, a very prolific advantage in other directions, as it would in her case; the only other vessels in the world of commensurate size and speed were the United States *Lexington* class, of which the protection appears to have been reduced to a dangerous degree. The similarity between the curves of the *Blonde* and *Newcastle*, in spite of the latter being of 35 per cent greater displacement, again exemplifies the value of length in vessels of this class. The comparisons which can be made amongst the various types are too numerous to permit of further elaboration.—"Shipbuilding and Shipping Record," December, 1922.

A NEW TYPE OF STUFFING-BOX.

There is little doubt that the ordinary type of gland and stuffing-box, familiar to every engineer ashore and afloat, leaves much to be desired, especially when the conditions are unfavorable. A good deal has been effected in the way of improvements, but most of the modifications have related to the nature of the packing itself rather than the means of application.

The device shown in the accompanying diagrams appears to represent a new departure, since the packing is fed through specially cut passages, and the pressure is applied through the medium of a tangentially arranged plunger and cap nut. A further feature is the provision of a discharge passage, sealed by a plug. If this plug be removed, fresh packing may be introduced through the plunger orifice, while the old packing finds an outlet through the discharge hole.



The Vygeboom stuffing-box. Packing is inserted through a tangential passage

Several important claims are made for the device by the makers, the Vygeboom General Export and Engineering Co., 43, Hoofdsteeg, Rotterdam. An outstanding recommendation is the possibility of feeding the stuffing-box with fresh packing during the running of the machine, and this point is worthy of consideration, particularly with reference to stern tubes. It is contended that the rotary motion of the shaft facilitates the operation, and the effect is partly the same with reciprocating shafts, although perhaps not

so great in extent. The packing chamber is tapered, and the pitch, together with the depth of the groove, may vary to suit different conditions; a range of stock sizes of packing is not required for a number of glands.

The form of construction illustrated is intended to show the main principles of the system, and may be varied according to requirements. In a large box, for example, several tangential passages may be provided. For the small spindles of fuel valves, fuel pumps, etc., in Diesel engines a more simple arrangement is adopted, while the makers claim that the Vygeboom type of box is also suited to such mechanism as piston-cooling, telescopic pipes, pumps, compressors, and the like.—"The Motor Ship."

SOME RECENT RESEARCHES ON LUBRICATION.*

By Dr. T. E. Stanton.

The Present Position of the Problem of Lubrication.—The theory of lubrication of Osborne Reynolds, published in 1886, was based on the existence between two lubricated surfaces, of a film of lubricant moving according to the laws of hydrodynamics, and was for long taken to afford a complete explanation of all the phenomena of lubrication. In recent years, however, the inadequacy of this theory to explain all the observed facts in the practice of lubrication has become more and more recognized. The main considerations which have led to a modification of the older view being as follows:—

(1) The discovery that lubrication can exist between surfaces which are so close together that no motion of the lubricant subject to the laws of hydrodynamics can take place between them.

(2) The relative superiority as lubricants of the fixed oils such as sperm, rape, and castor, over mineral oils of the same viscosity.

(3) The marked increase in the efficiency of worm gears, obtained by the addition of small quantities of fatty acids to mineral oils, the viscosity being very little affected by the addition.

(4) The well-known advantage obtained by the use of certain bearing metals.

All these facts appear to be consistent with the existence of a thin film of lubricant of molecular dimensions separating the surfaces, in which case the resistance must depend on the chemical constitution of both the lubricant and the lubricated surfaces. This kind of lubrication has been called "boundary lubrication." Lubrication of the Reynolds' type is here referred to as "perfect" lubrication.

The Range of the Proposed Investigation.—On making a careful examination of the available evidence, it appeared to the writer that there were two directions in which fresh light on the problem might be sought.

In the first place it appeared to be very desirable that the theoretical effect of the use of lubricants of different viscosity in lubrication of the Reynolds' type should be more generally recognized by engineers than is the case at present. Secondly, an extension of the previous experimental work appeared to be very desirable in order to decide whether lubrication of the Reynolds' type was of common occurrence in engineering practice, or, as

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* Institution of Mechanical Engineers, Ninth Thomas Hawksley Lecture (abstract), delivered in London, December 1.

some engineers are inclined to think, a somewhat rare phenomenon found to exist under the special conditions of the Michell thrust bearing and the bath lubricated journals used by Tower.

Review of the Theory.—The work of Tower, carried out in 1883-1885, first led mathematicians to take up the study of lubrication as a particular problem in fluid motion. Tower discovered that the lubricant at the crown of the bearing was under considerable pressure, and by connecting small holes at different points of the bearing surface to a Bourdon pressure guage he was able to obtain a rough estimate of the difference of pressure. From the fact that a summation of the vertical components of the oil pressure was approximately equal to the total load, he was able to demonstrate that the two surfaces of bearing and journal were completely separated by an oil film.

This discovery led Osborne Reynolds to take up the problem from the theoretical side and to study the conditions under which by the laws of hydrodynamics a separating film could be permanently maintained between the surfaces such that the pressure distribution in the film would account for the load and the shear stress distribution would account for the frictional resistance actually experienced.

Such a state of motion, he found, could only exist when the following conditions were satisfied:—

(1) That there must be a finite difference between the values of the radii of the journal and the bearing, and that the bearing must be in general excentric to the journal.

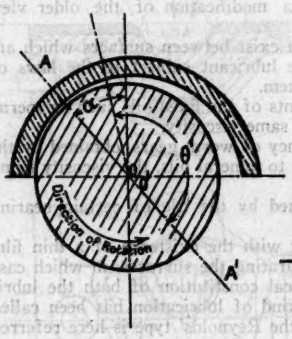


FIG. 1

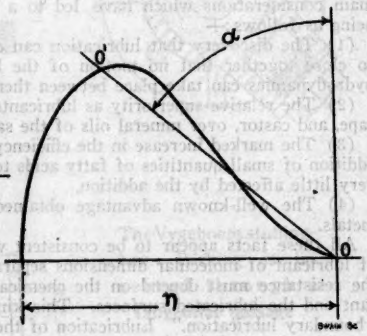


FIG. 2

(2) That the attitude of the bearing relative to the journal would be as shown in Fig. 1. Thus O is the center of the journal and O′ that of the bearing. Then evidently the line O O′ produced passes through A, the point of nearest approach of the surfaces. If η be the difference of the radii of the two surfaces, it is evident that O O′ cannot exceed η, and that its value must lie between zero when the surfaces are concentric, and η when the surfaces are touching.

It is usual to write $O\,O' = c\,\eta$ and to call c the excentricity, c having any value between O and unity. The angle α which O A makes with the vertical may be called the attitude of the bearing, and when the attitude and excentricity are specified the position of the bearing is fixed.

Under such conditions Reynolds showed that from the laws of hydrodynamics the pressure changes in the film must be given by the equation—

$$\frac{dp}{d\theta} = \frac{6\mu U a c}{\eta^2}\left(\frac{\cos\theta - \cos\theta'}{(1 + c\cos\theta)^3}\right)$$

where θ is measured from O A′ in the direction of rotation of the journal, μ is the coefficient of viscosity of the lubricant, U is the relative speed of the two surfaces, a the radius of the journal, and θ' the value of θ when the pressure is a maximum.

This is the fundamental equation of lubrication from which all succeeding investigators, Michel Petroff, Sommerfeld and Harrison have proceeded. The obvious method of finding whether in the case of Tower's experiments a hydrodynamical state of flow existed, was to integrate equation (1) and compare the pressure so calculated with that actually observed by Tower. Unfortunately this comparison is a matter of extreme difficuty, not on account of the integration itself, but because before the numerical value of p corresponding to any particular value of θ can be calculated, it is necessary to know the attitude, the excentricity, the viscosity of the oil, the radius difference of the bearing and journal, and the angular position of the point of maximum pressure.

The Complete Solution of the Half Bearing—The difficulties encountered by Reynolds in dealing with the theoretical case of a half bearing in which, owing to high load or slow speed, the distance between the surfaces becomes less than half the difference of the radii of the bearing and the journal, were eventually overcome by Sommerfeld in 1900. Without the aid of advanced mathematics he succeeded in obtaining a solution of the problem of the half bearing for all values of the excentricity. This solution is not entirely applicable to practical conditions in that the falling off of the pressure at the ends of the journal is neglected.

One of the most interesting and valuable of Sommerfeld's results is the relation which he obtained between the attitude and excentricity of a half bearing and what must hold whatever the load, speed and lubricant employed. This relation is shown in Fig. 2, in which the radius vector of the curve gives the displacement of O′ from O and its angular position the attitude of the bearing. It will be seen that, as previously shown by Osborne Reynolds, there are two values of the excentricity for each value of the angular position of the point of nearest approach.

Application of the Theory to a Practical Case.—A still more important application of Sommerfeld's work is the light which is thrown on the effect on the frictional resistance of changes in the lubricant employed in the case of heavily loaded bearings.

In order to show this I have worked out from Sommerfeld's equations the frictional resistances of a 4-inch bearing in which the load and speed are always the same and the lubricant alone is varied.

The results are given in the following table:—

TABLE I.

Diameter of journal	4 in.
Diameter of bearing	4.004 in.
Speed	188 revolutions per minute
Load	3,420 lb. per square inch run

Viscosity of lubricant C.G.S.units.	Attitude of bearing.	Excentricity of bearing.	Nearest approach in inches.	Coefficient of friction
	Degrees.			
0.223	129	0.94	6×10^{-5}	0.00094
0.488	135	0.82	18×10^{-5}	0.00096
0.824	133	0.65	35×10^{-5}	0.00099

It will be seen that an increase in the viscosity of the lubricant in the ratio of about 4 to 1 has quite a small effect on the frictional resistance the increase in which is of the order of 5 per cent. The reason of the comparatively small effect on the friction is, as will be seen, due to the fact that with the more viscous oil the surfaces are kept much further apart than with the thinner oils and that consequently the speed remaining the same the product of the coefficient of viscosity into the space rate of shear, which is the frictional resistance, remains nearly constant. This example is a very instructive one in that it clearly demonstrates the insufficiency of the deduction that observations showing that friction is independent of viscosity must be taken as conclusive evidence of the existence of boundary lubrication.

The Lubrication of Flat Surfaces.—It is of interest to note that whatever limitations may be found in practice to the existence of a film of lubricant of the Reynolds' type in cylindrical lubrication, difficulties in the setting up of this condition rarely appear in the case of flat surfaces, for which the theory has been worked out by Michell. In this case, by eliminating the curvature of the surfaces, Michell has been able to obtain the pressure distribution in two dimensions and thus to take account of the leakage at the sides of the plate.

The Experimental Investigation of the Characteristics of Cylindrical Journal Lubrication Under Limiting Conditions of Pressure.—In considering the details of the apparatus to be designed for the experiments at Teddington, the first consideration was whether, in view of the difficulties which Reynolds had encountered in the interpretation of Tower's results, some of the unknown factors could not be evaluated experimentally. One of these unknown factors in Tower's experiments was the temperature of the oil film. On examination of the possibility, in the present tests, of a direct observation of the temperature of the film, it appeared probable that this could be measured with sufficient accuracy by thermoelectric methods, and it was decided to make provision for this in the design.

A still more valuable asset in the interpretation of the experimental results, if it were possible to make a determination of its value, would be the difference of radii between the bearing and the journal. In 1886 high accuracy in the measurement of the internal diameter of cylinders had not been developed, and it is not improbable that with the best methods available at that time the limits of accuracy of the determination would have been $\pm$ 15 per cent. With the aid of modern methods, however, an accuracy of $\pm$ 1 per cent in the measurement of the radius difference could be hoped for, so that as far as the dimensions of the bearing in its unstrained state are concerned the difficulty can be overcome.

It was realized, however, by Osborne Reynolds, that there were two factors in the conditions of running which would materially affect the curvature of the bearing, one the expansion of the metal of the bearing due to the rise of temperature caused by the frictional resistance, and the other

the elastic distortion of the bearing under the imposed load, with the result that the value of the radius difference would vary both with the load and the speed.

The amount of these distortions was actually calculated from Tower's results, and for his 4-inch bearing was found to involve increases in the radius difference of about 50 per cent due to an increase of load from 100 to 400 pounds per square inch mean pressure, and 25 per cent due to an increase of speed from 100 to 450 revolutions per minute. In order to obviate this difficulty it is necessary to increase the rigidity of the bearing and to work with a much larger radius difference so that the effect on the latter of any distortion due to load and temperature is comparatively small.

General Arrangement of the Apparatus.—This is shown in Figs. 3 and 4. The bush, which is of phosphor bronze ground and lapped out to the required diameter is supported in a steel housing by means of ball bearings, thus affording means of independent rotation for the purpose of the pressure observations. The loading is obtained by cast iron flitch plates bolted to the sides of the housing. A small oil dash-pot is provided to damp out any oscillations of the bearing, and a scale pan and calibrated spring for measuring the frictional torque. A special feature is the load relieving device by means of which the load is gradually taken off the journal before stopping its rotation, and for the gradual imposition of the load in starting up. The journal which is made of tool steel hardened and ground was overhung and held by a chuck at the end of the driving shaft. The Bourdon pressure gauge for measuring the oil pressure is carried by the steel housing of the bearing and is connected to the pressure hole by soft copper tubing, which could be bent into any desired position when rotating the bearing into the different angular positions of the pressure hole. For the measurement of the temperature of the film an iron-constantan thermo-junction was inserted into a small hole drilled in the crown of the bearing, and in such a position that the junction itself was in contact with the surface of the revolving journal.

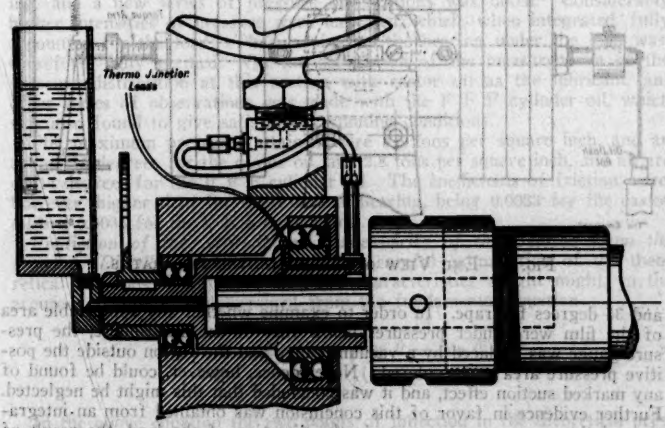


FIG. 3.—SECTION OF JOURNAL AND BEARING IN EXPERIMENTAL APPARATUS.

Results of the Experiments.—The early experiments on pressure distribution were made with bearings in which the radius difference was about 0.1 per cent of the radius of the journal. It was realized that under these conditions the effect of temperature and elasticity would probably be such that no reliable estimate of the attitude and distance apart of the surfaces could be made, but it was thought worth while to make them for the sake of obtaining a check on the previous experimental work under similar conditions. On making a series of pressure observations, curves of pressure distribution were obtained from which the resultant vertical component of the oil pressure could be determined by integration. The values of these components were found to be appreciably in excess of the mean load per inch run of the bearing as they should be in view of the falling off of the intensity of pressure at the ends of the bearing. It was evident, therefore, that the perfect state of lubrication obtained by Tower in his oil bath experiments could be easily reproduced in the present apparatus. As these, however, were not the conditions under which the problem was to be studied, the work with the high values of the radius difference was proceeded with, and attention was finally concentrated on a series of experiments with a bearing in which the radius difference was 2 per cent.

The pressure distribution over the central section of this bearing, under a total load of 690 pounds and at a journal speed of 1,000 revolutions per minute, is shown in Fig. 5, for the case of sperm oil. As the main object of the experiments was the determination of the attitude and excentricity of the bearing no attempt was made to investigate the extent to which the pressure fell off, in a longitudinal section, towards the ends of the bearing.

The arc of contact of the film over which pressures above atmospheric existed is somewhat restricted, the extent being 30 degrees for sperm oil

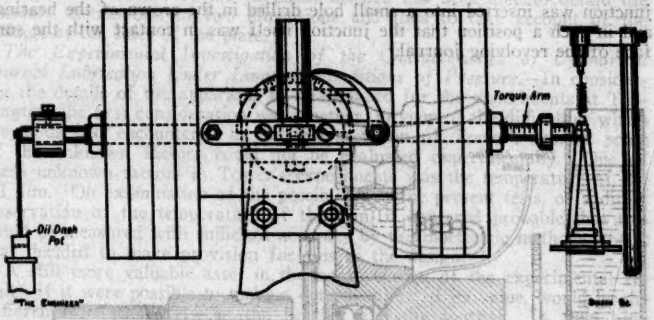


FIG. 4.—END VIEW OF EXPERIMENTAL APPARATUS.

and 35 degrees for rape. In order to examine whether any appreciable area of the film were under pressures below that of the atmosphere, the pressure gauge was replaced by a vacuum gauge and the region outside the positive pressure area was explored. No evidence, however, could be found of any marked suction effect, and it was concluded that this might be neglected. Further evidence in favor of this conclusion was obtained from an integration of the pressure components in the direction of the load, the result of which, allowing for the falling-off of pressure at the ends of the bearing,

was approximately equal to the total load. The values of the observed frictional torques were 0.25 pound-inch for sperm oil and 0.59 pound-inch for rape, giving values of the normal coefficient of friction 0.00072 and 0.0017, respectively.

The results of these experiments were so satisfactory that an attempt was made to obtain still higher intensities of oil pressure by a further increase of the radius of the bearing.

For this purpose a phosphor bronze bush of 1.06 inches internal diameter was prepared. Tests were made with this bearing in the same manner and under the same conditions of load and speed as before. As these conditions were severe it was decided to use the best known lubricant for high pressures, and a trial was made with castor oil. It was found that a steady state of lubrication could be set up with this oil and the pressure distribution was determined. This appeared to be quite satisfactory, but it was noticed that although the arc of contact was very much reduced, the intensities of the pressures, although considerably higher than in the case of the 1.02-inch bearing, were not sufficient, when their integration was carried out, to account for the total load.

It was thought at first that this was due to some irregularities in the surface of the bearing, and that the discrepancy would disappear after a period of "running in." No improvement, however, was found after a prolonged period of running, and it became evident that this was not the source of the difficulty. The general contour of the pressure curve obtained was so similar to that previously obtained for the first bearing that any explanation based on a change in the lubricating conditions did not seem probable, and the possibility of the central section of the bearing not being that of maximum pressure was examined. Attention was drawn to the fact that the bearing—see Fig. 3—was unsymmetrically supported in the housing by means of the large and small ball bearings, and the possibility of a deformation of the bearing on the two point support thus provided seemed feasible. In order to test this experimentally, a new pressure hole was made in the section of the bearing directly under the main ball bearing, and a new series of pressure observations was taken. Considerably higher intensities of pressure were indicated, which, when integrated, fully accounted for the load. The warping of the bearing under the load was, therefore, fully verified. On the completion of the measurements of the pressure distribution at this section with castor oil as the lubricant, another series of observations was made with the F F F cylinder oil, which was also found to give satisfactory running conditions.

The maximum pressures attained were 3.5 tons per square inch, and an arc of 15 degrees for the castor oil, and 2.8 tons per square inch, and an arc of 14 degrees for the F F F cylinder oil. The coefficients of friction were, however, higher than for the 1.02-inch bearing, being 0.0023 for the castor oil and 0.0035 for the F F F cylinder oil.

Prediction of the Attitude and Excentricity of the Bearing from the Pressure Distribution Curves.—After a careful examination of the theoretical relations, it appeared that the characteristics sought might, to the accuracy required, be determined from the fundamental equation—

$$\frac{dp}{d\theta} = \frac{6\mu U a c}{\eta^2}\left(\frac{\cos\theta - \cos\theta_1}{(1 + c\cos\theta)^3}\right)$$

In the above equation the two points of inflection in the theoretical pressure distribution curve are given by

$$\theta = 180 \text{ deg. and } \cos\theta = \frac{1 + 3\cos\theta^1}{2c}.$$

The point corresponding with $\theta = 180$, is evidently the point of nearest approach on the off side of the bearing, and it is clear, therefore, that if from the actual pressure distribution curve the point of inflection on the off side could be detected the attitude of the bearing as defined by the angle α would have been determined.

From an examination of the curve—Fig. 5—it will be seen that although this point cannot be located by eye with great precision, the regularity of the observations is such that in all probability it can be located to an accuracy of a fraction of a degree.

The possibility of obtaining an approximate value of the attitude of the bearing from an inspection of the actual pressure distribution curve, suggested the following method of calculating the characteristics of the bearing.

The observed pressures having been plotted on an angle base of which the zero was arbitrary, the position of the point of inflection on the off side was estimated and its angular position was scaled off. In the case of the experiment on sperm oil—Fig. 5—this is seen to be 6 degrees. This point, therefore, on the scale of θ, corresponds with 180 degrees. The angular position of the point of maximum pressure is seen to be 2 degrees 45 minutes behind the point of nearest approach. The angular position of the point of maximum pressure on the scale of θ is, therefore, 180 degrees—2 degrees 45 minutes, or 177 degrees 15 minutes. This gives cos θ^1 in the fundamental equation. Therefore, given any two values of $\frac{dp}{d\theta}$ at positions θ_1 and θ_2, we have two simultaneous equations from which the unknown quantities μ aand c can be obtained. These values θ_1 and θ_2 are quite arbitrary, but for convenience they have been chosen as 180 degrees, the position of the point of nearest approach, and a corresponding point on the "on" side.

In the case of the sperm oil curve—Fig. 5—the scaled values of $\frac{dp}{d\theta}$ at $\theta = 180$ degrees and 170 degrees 30 minutes were found to be $-52{,}700$ and $+13{,}200$ pounds per square inch per radian, respectively.

Substituting these values in the fundamental equation:

$$c = 0.9946;$$
$$\mu = 0.32 \text{ (in C.G.S. units)}.$$

At a temperature of 35.8 degrees Cent., which was the measured temperature of the film the value of μ for sperm oil at atmospheric pressure is 0.25. It would appear, therefore, that if the above calculated value of $\mu = 0.32$, is the correct one, the pressure in the oil film must have been such as to raise the viscosity of the oil by about 25 per cent above its atmospheric pressure value.

The observed maximum pressure of the oil was 1.44 tons per square inch, and if we refer to the curves of variation of viscosity with pressure obtained by Mr. Hyde, we find that such an increase is about what would be expected.

The value of $c = 0.9946$ would appear, therefore, to be approximately correct, and the minimum distance apart of the surfaces can be calculated. This is given by $h = \eta(1 - c)$, and is in this case 0.000054 inch. A similar calculation from the pressure distribution in the rape oil experiments with the same bearing gave

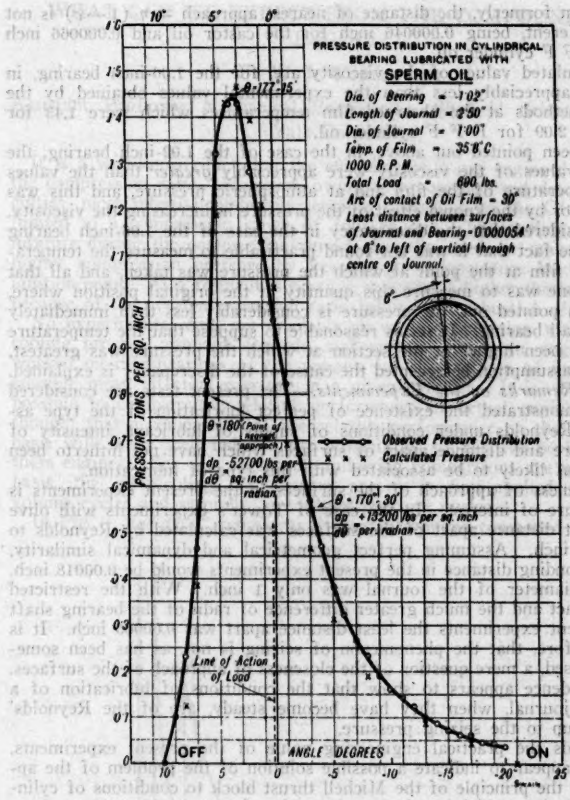

PRESSURE DISTRIBUTION IN CYLINDRICAL BEARING LUBRICATED WITH SPERM OIL
Dia. of Bearing = 1·02"
Length of Journal = 2·50"
Dia. of Journal = 1·00"
Temp. of Film = 35·8° C
1000 R. P. M.
Total Load = 690 lbs.
Arc of contact of Oil Film = 30°
Least distance between surfaces of Journal and Bearing = 0·000054" at 6° to left of vertical through centre of Journal.
θ = 177° 15'
θ = 180° (Point of nearest approach)
dp/dθ = -52700 lbs per sq. inch per radian
θ = 170° 30'
dp/dθ = +13200 lbs per sq. inch per radian
Observed Pressure Distribution
Calculated Pressure
Line of Action of Load
PRESSURE TONS PER SQ. INCH
ANGLE DEGREES
OFF
ON
10° 5° 0° 5° 10° 15° 20° 25°
Scale

FIG. 5.

$c = 0.9904$,

or, least distance apart of surfaces = 0.000096 in.

Following the same procedure in the case of the results obtained with the 1.06-inch bearing, the values of the viscosity and excentricity were found to be:—

For the castor oil

$$\mu = 1.14, \quad c = 0.99845.$$

For the F F F cylinder oil

$$\mu = 1.87. \quad c = 0.9978.$$

The values of c in both cases are considerably higher than those for the 1.02-inch bearing, but since the radius difference η is now three times

greater than formerly, the distance of nearest approach $=\eta\ (1-c)$ is not greatly different, being 0.000046 inch for the castor oil and 0.000066 inch for the F F F cylinder oil.

The calculated values of the viscosity are, for the 1.06-inch bearing, in each case appreciably less than the experimental values obtained by the ordinary methods at the observed film temperatures which were 1.43 for castor and 2.09 for F F F cylinder oil.

As has been pointed out above, in the case of the 1.02-inch bearing, the calculated values of the viscosity were appreciably *greater* than the values at the temperature of the film and at atmospheric pressure, and this was accounted for by the known effect of the pressure in increasing the viscosity.

It is considered that the discrepancy in the case of the 1.06-inch bearing is due to the fact that it was not found practicable to measure the temperature of the film at the point at which the pressure was taken, and all that could be done was to measure this quantity at the original position where, as has been pointed out, the pressure is considerably less than immediately under the ball bearing. It seems reasonable to suppose that the temperature would have been higher at the section at which the pressure was greatest, and if this assumption be accepted the cause of the discrepancy is explained.

General Remarks on the Experiments.—The present tests are considered to have demonstrated the existence of perfect lubrication of the type assumed by Reynolds under conditions of supply of lubricant, intensity of fluid pressure and distance apart of surfaces which have not hitherto been considered as likely to be associated with this type of lubrication.

The closeness of approach of the surfaces in the present experiments is also a feature of interest. In the case of Tower's experiments with olive oil the least distance apart of the surfaces was calculated by Reynolds to be 0.00072 inch. Assuming perfect geometrical and dynamical similarity, the corresponding distance in the present experiments would be 0.00018 inch, since the diameter of the journal was only 1 inch. With the restricted arc of contact and the much greater difference of radii of the bearing shaft in the present experiments the least distance apart was 0.00005 inch. It is clear, therefore, that the phenomenon of seizing is not, as has been sometimes supposed, a mere question of the closeness of approach of the surfaces. All the evidence appears to show that the conditions of lubrication of a cylindrical journal, when they have become steady, are of the Reynolds' type right up to the seizing pressure.

As regards the practical engineering value of the present experiments, the results appear to indicate a possible solution of the problem of the application of the principle of the Michell thrust block to conditions of cylindrical lubrication, which, as far as the author is aware, has not yet been solved.

It is fully recognized by designers of internal combustion engines that the chief problem which confronts them in the production of higher piston speeds and consequently greater efficiency is the "big end" problem, *i.e.*, the limit of speed is defined by the limiting pressure on the big end bearing of the connecting-rod. From the success which has been attained in the present experiments in the establishment of very high bearing pressures with arcs of contact of the order of 20 degrees, it would appear that a cylindrical bearing constructed of a number of such bearing blocks, each capable of independent adjustment, might prove to be a successful means of obtaining higher pressures in big end bearings than are at present attainable.—"The Engineer," December, 1922.

WHAT CAN WE EXPECT FROM THE COMPOUND GAS ENGINE?

By W. J. Wohlenberg,

Assistant Professor Mechanical Engineering, Sheffield Scientific School, Yale University.

It is a well-known fact that the efficiency of a gas engine depends upon the compression ratio. However, owing to preignition and detonation difficulties, most internal-combustion engines operating on the Otto cycle are limited to a compression range of four to one. The cooling of the gas mixture during the compression period may be used to make possible a higher compression pressure by keeping the temperature below the preignition point. As a result it is possible to realize a higher thermal efficiency because of the greater expansion ratio following combustion of the charge.

No experimental data are available on engines operating with the intercooling process, although Ricardo and certain European engineers have worked on modifications of the process for the purpose of increasing the possible mean effective pressure in a given cylinder. It is, therefore, not possible to compare the relative performance of actual engines, but it is thought that a comparison of the relative performance on the air-standard basis will fairly well indicate the possibilities of relative performance of these engines. For this reason all following comparisons are made on this basis, the intercooling compression process is first applied to constant-volume cycles and secondly to constant-pressure cycles.

In Fig. 1, *dad'a'b'C'* represents a constant-volume compound intercooling

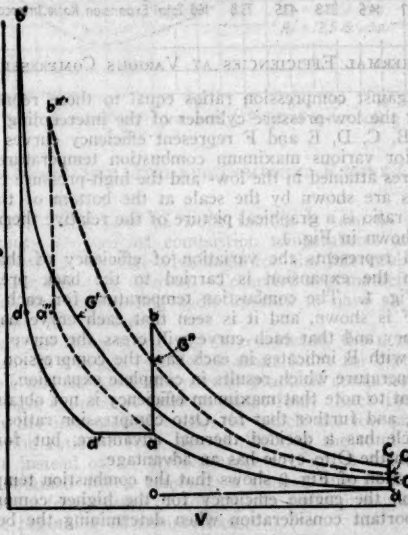


Fig. 1.—Diagrams of Compound and Otto-cycle Engines.

cycle and *dab*C and *da"b"*C" respectively represent Otto single compression cycles in which the compression pressure is carried, first, to that in the low-pressure cylinder of the intercooling cycle and, secondly, to that existing in the high-pressure cylinder of the intercooling cycle. An inspection of the diagrams at once shows the very marked increase of the expansion ratio because of the intercooling process. The gain in efficiency due to this is partly offset by the heat rejected during the cooling process *ad'*.

Referring to Fig. 2, Curve A represents the thermal efficiency of the Otto

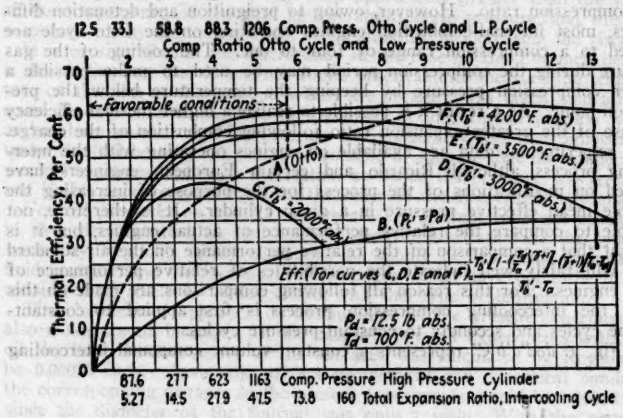

12.5 33.1 58.8 88.3 120.6 Comp. Press. Otto Cycle and L. P. Cycle
Comp Ratio Otto Cycle and Low Pressure Cycle
2 3 4 5 6 7 8 9 10 11 12 13
70
60
50
40
30
20
10
0
Thermal Efficiency, Per Cent
←Favorable conditions→
F. (T_b' = 4200° F. abs.)
E. (T_b' = 3500° F. abs.)
D. (T_b' = 3000° F. abs.)
A (Otto)
C. (T_b' = 2000° F. abs.)
B. (P_c = P_d)
Eff. (For curves C, D, E and F).
P_d = 12.5 lb abs.
T_d = 700° F. abs.
87.6 217 623 1163 Comp. Pressure High Pressure Cylinder
5.27 14.5 27.9 47.5 73.8 160 Total Expansion Ratio, Intercooling Cycle

FIG. 2.—THERMAL EFFICIENCIES AT VARIOUS COMPRESSION RATIOS.

cycle plotted against compression ratios equal to those represented on the upper scale for the low-pressure cylinder of the intercooling engine.

The curves B, C, D, E and F represent efficiency curves for the intercooling cycle for various maximum combustion temperatures. The compression pressures attained in the low- and the high-pressure cylinders under these conditions are shown by the scale at the bottom of the chart. The total expansion ratio is a graphical picture of the relative thermal efficiencies of the cycles shown in Fig. 1.

The curve B represents the variation of efficiency of that intercooling cycle in which the expansion is carried to the back pressure, whence $Pc = Pd$ in Fig. 1. The combustion temperature for each of the curves C, D, E and F is shown, and it is seen that each curve has a maximum point of efficiency and that each curve will cross the curve B. The point of intersection with B indicates in each case the compression ratio for this combustion temperature which results in complete expansion.

It is important to note that maximum efficiency is not obtained with complete expansion, and further that for Otto compression ratios below six the intercooling cycle has a decided thermal advantage, but for compression ratios over seven the Otto cycle has an advantage.

Further inspection of Fig. 2 shows that the combustion temperature has a greater effect on the engine efficiency for the higher compression ratios. This is an important consideration when determining the best method of

governing the engine. It is apparent that to maintain a high efficiency at light loads for high compression ratios, the engine should be so governed as to maintain the combustion temperature nearly constant.

It will now be interesting to determine the probable effect on the engine weight by using the intercooling process within the range of conditions in which it possesses a thermal advantage.

The ratio of work obtained from the compressed intercooling and from the Otto-cycle engine is shown in Fig. 3 for different compression ratios and

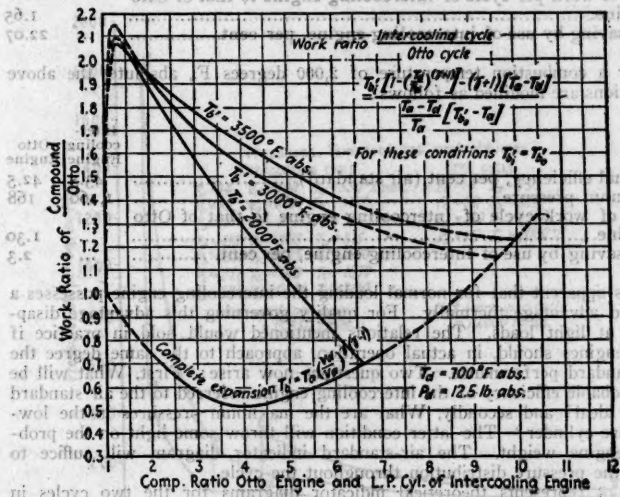

Work ratio Intercooling cycle / Otto cycle
For these conditions
Work Ratio of Compound Otto
Comp. Ratio Otto Engine and L.P. Cyl. of Intercooling Engine
Complete expansion
Pd = 12.5 lb. abs.

FIG 3.—COMPARATIVE OTTO AND COMPOUND ENGINE OUTPUTS.

final temperatures. The ratio of work factors is seen to be considerably greater than unity for normal combustion temperatures and compression ratios, which shows a thermal advantage for the intercooling cycle (that is, for compression ratios up to five). To arrive at some definite figures for a case in which the intercooling cycle appears to have an advantage, the following example will suffice:

If a fuel such as illuminating gas is considered, the normal practicable compression ratio in a single cylinder is about four; for certain liquid fuels such as gasoline and kerosene the normal practical compression of the fuel mixture in a single cylinder is likewise about four. Such fuels may be used successfully in the intercooling engine in which the low-pressure cylinder has a compression ratio of four, but in this case the total expansion ratio will now be 28 instead of 4. By reference to Figs. 2 and 3 the following data may be obtained for a combustion temperature of 3,500 degrees F. absolute, and for a temperature of 700 degrees and a pressure of 12.5 pounds absolute at the beginning of compression.

	Intercooling Engine	Otto Engine
Compression ratio (l. p. cyl.)	4	4
Expansion ratio (total)	27.9	4
Thermal efficiency, per cent (air standard)	55	42.5
Compression pressure, lb. (l. p. cyl.)	88.3	88.3
(h. p. cyl.)	623	...
Temperature after compression, deg. F. absolute	1,048	1,048
Maximum pressure, lb. per sq. in	2,050	295
Ratio of work per cycle of intercooling engine to that of Otto engine	...	1.65
Fuel saving by use of intercooling engine, per cent	...	22.07

For a combustion temperature of 2,000 degrees F., absolute, the above conditions are modified as follows:

	Intercooling Engine	Otto Engine
Thermal efficiency, per cent (air standard)	43.5	42.5
Maximum pressure	1,160	168
Ratio of work cycle of intercooling engine to that of Otto engine	...	1.30
Fuel saving by use of intercooling engine, per cent	...	2.3

It is apparent that for normal loading the intercooling engine possesses a decided advantage thermally. For quality governing this advantage disappears at light loads. The relations mentioned would hold in practice if both engines should, in actual operation, approach to the same degree the air-standard performance. Two questions now arise: First, What will be the probable efficiency of the intercooling engine referred to the air standard as an ideal? and secondly, What are the maximum pressures in the low-pressure cylinder? The latter condition will throw some light on the probable engine weight. The air-standard indicator diagram will suffice to show the pressure distribution throughout the cycle.

Fig. 4 represents theoretical indicator diagrams for the two cycles in which a compression pressure of 80 pounds is used in the first stage of the intercooling cycle and the total compression pressure for the Otto cycle is likewise 80 pounds.

The volumes of the low-pressure intercooling cycle cylinder and that of the Otto cycle are each taken at one cubic foot and the temperature and pressure at the beginning of compression are the same in both cases. Therefore, the weight of the working fluid is the same in both cases, and since the same temperature rise is used during each combustion period, the same quantity of heat is added per cycle, the specific heat being practically independent of the pressure for the conditions given.

Under these conditions 53 per cent more work is accomplished per cycle by the intercooling engine and the ideal thermal efficiency of the cycle is 58 per cent.

A close inspection of these diagrams will, as previously stated, lead to some information concerning the probable relative weights of the two engines. Referring to the expansion curve $b'C'$, it is seen that the pressure at the point G', the opening of the transfer valve, is approximately 280 pounds per square inch absolute and the temperature 2,120 degrees F. The low-pressure cylinder is therefore exposed to maximum pressures somewhat less than those existing in the Otto engine cylinder. It will certainly not

weigh any more than the Otto engine cylinder. The high-pressure cylinder is exposed to very high pressures, but its volume is only 17 per cent that of the low-pressure cylinder. By means of the addition of the intercooler, high-pressure cylinder, high-pressure piston and connecting rod, 53 per cent more work is accomplished per cycle. It seems improbable that the weight of these additional parts will serve to offset the gain in power. To answer the first question mentioned, namely, What is the probable relative performance of the two cycles referred to the air standard as an ideal? will require an analysis of the engine losses.

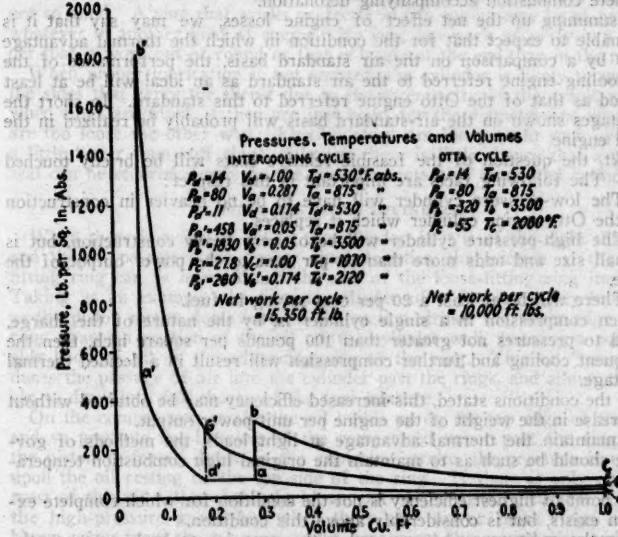

Pressures, Temperatures and Volumes
INTERCOOLING CYCLE
OTTA CYCLE
Net work per cycle = 15,350 ft lb
Net work per cycle = 10,000 ft. lbs.
Pressure, Lb. per Sq. In. Abs.
Volume Cu. Ft.

FIG. 4.—DIAGRAMS SHOWING THE THEORETICAL WORK PER CYCLE.

Among the engine losses may be mentioned jacket losses, leakage, friction, expansion losses and the possibilities of poor combustion, or combustion losses.

The cylinder or jacket losses probably will be less in the intercooling cycle because the maximum temperature of the charge is no higher and during combustion and early expansion periods the gases are contained in a small cylinder in which the space occupied will be of such shape as to have probably less area per unit volume than would be the case in the Otto-cycle engine for corresponding events. Leakage would be greater in the intercooling engine because of the higher pressures. In the present state of the art, however, it should be possible to reduce the leakage losses, even for the higher pressures, to a very small fraction of the total. Frictional losses appear to be about the same for both cases.

Under combustion losses would be included those resulting from condensation of fuel vapors because the intercooling process may have been carried

too far. Any such condensation will, obviously, increase the combustion losses. This would have to be guarded against by so regulating the intercooling that the dew point is not reached. If this precaution is observed, the combustion process in the intercooling engine will be at least as good as that in the Otto engine.

DETONATION MAY BE AN ADVANTAGE.

Detonation, if it should occur and cannot be avoided, may prove an actual advantage, as there is certainly a thermal advantage in having the early complete combustion accompanying detonation.

In summing up the net effect of engine losses, we may say that it is reasonable to expect that for the condition in which the thermal advantage exists by a comparison on the air standard basis, the performance of the intercooling engine referred to the air standard as an ideal will be at least as good as that of the Otto engine referred to this standard. In short the advantages shown on the air-standard basis will probably be realized in the actual engine.

Next, the question of the feasible relative costs will be briefly touched upon. The following facts are important in this respect:

1. The low-pressure cylinder will have to be no heavier in construction than the Otto-engine cylinder which it replaces.

2. The high-pressure cylinder will be of very heavy construction, but is of small size and adds more than 50 per cent to the power output of the engine.

3. There will be at least a 20 per cent saving in fuel.

When compression in a single cylinder is, by the nature of the charge, limited to pressures not greater than 100 pounds per square inch, then the subsequent cooling and further compression will result in a decided thermal advantage.

For the conditions stated, this increased efficiency may be obtained without an increase in the weight of the engine per unit power output.

To maintain the thermal advantage at light loads, the methods of governing should be such as to maintain the original high combustion temperature.

The point of highest efficiency is not the condition for which complete expansion exists, but is considerably above this condition.

Since the engine may be operated with expansion to back pressure, the exhaust will be noiseless under these conditions.

The intercooling process is at a thermal and power disadvantage when it is not the means of increasing the final compression pressure.—"Power."

HOLDING THE PISTON SEAL IN A GAS ENGINE.

By W. F. Osborne, Supervisor Manufacturers' Service, Texas Company.

Piston rings are put on a piston to help it hold the gases within the cylinder until their work is done. If a plain piston were used without rings, the unburned fuel mixture would leak past the piston on the compression stroke, wasting good fuel and lowering the maximum compression available. On the explosion stroke a part of the burned gases would leak past the piston and again power would be lost. Because of its length it is practically impossible to fit the entire piston tight enough to eliminate the leak-

age of gas, and even if it were feasible, the greater friction of the closely fitting piston would eat up the power saved.

Putting rings on the piston allows it to have a looser fit in the cylinder, thereby lowering its friction, without there being any loss of power. Rings that fit tightly against the cylinder walls will allow less gas leakage than loosely fitting rings, but their friction will be high. If the rings are too loose, they will not properly prevent gas leakage. The proper ring will balance up the question of power loss by leakage and the power loss by high ring friction.

There are all kinds of rings on the market; some are good, and some are not so good. A ring that works perfectly in one engine may be of no value whatever in another kind of engine. No ring can work properly in a cylinder that is worn out of round or that is larger at the end than in the middle.

When proper lubrication is not secured, the cylinder is likely to wear on the sides that carry the thrust due to the eccentricity of the crank, so that when rings are fitted, if they are just right for the diameter one way, they are too loose the other way. Flexible rings are likely to fit such cylinders a little better than stiff rings, although there is little chance that a perfect seal can be secured simply by the metallic contact of rings and cylinder.

RINGS FOR WORN CYLINDERS.

When there is a film of oil on the cylinder, if it is of the right thickness and the oil possesses the proper characteristics, the friction of the tight-fitting ring can be lessened and the seal of the loose-fitting ring improved. Taking as an example, the vertical cylinder of a four-stroke-cycle engine, as the piston moves downward on the suction stroke, the lower edge of the ring scrapes a portion of the oil off the cylinder wall, forming a considerable body of oil on the advancing side of the ring. This oil "wave" reduces the passage of air into the cylinder past the rings, and allows the piston to draw in a full charge of the fuel mixture.

On the compression stroke the oil builds up on the top edge of the ring, and so prevents the blowing of the compressed vapor past the rings. As the piston is forced down on the explosion stroke, the pressure is exerted upon the oil resting on the top side of the ring. If there is only one ring, most of this oil will probably be blown out of the piston clearance space by the high-pressure gases, but when there are two or more rings, any oil blown away from the top ring, packs up against the second ring and tends to prevent the further passage of the gas.

Another place for gas leakage is through the clearance space back of the rings, between the ring and its groove. If the ring is a loose fit in the groove particularly with respect to width, the vapor will find its way around the ring and power will be lost. The oil is of benefit here, too, as it fills this clearance space, and its resistance helps to prevent the gas leakage. The value of the oil in assisting the rings is particularly noticeable where the cylinder is not perfectly round, so that it is impossible for the rings to prevent leakage of gases without the assistance of the oil.

EFFECT OF VISCOSITY.

The physical properties of the oil that influence its seal-forming value are viscosity, surface tension with respect to metal, and cohesion. There may also be some chemical reaction between certain constituents of some kinds of oils and the metal of the cylinder and the piston which increases its adhesion to the metal and prevents the oil from being scraped off.

Other things being equal, we would expect that a high viscosity would cause the oil to offer a greater resistance to being blown out of the clear-

ance space and would also permit the accumulation of a larger body of oil on the advancing side of the rings than could be secured by a lower-viscosity oil. We find by actual experience that the piston seal can be greatly improved in many instances by raising the viscosity of the oil; better compression is secured and more power developed by the engine at a lower consumption of fuel per horsepower. This is especially true where the piston, cylinder and rings have become worn.

From this statement we might reach the conclusion that if a high-viscosity oil is good, a higher viscosity would be better. This is not true, as we sometimes find that if the viscosity is too high the oil will not work its way between the piston and the cylinder fast enough, the cylinder becomes dry, and many troubles follow the lack of sufficient oil.

CHOICE OF OILS.

Should the viscosity be a little low, there is not likely to be lack of lubrication, but simply loss of power through gas leakage and troubles from excessive oil in the cylinder and carbon deposits with fouled ignition points. So we must get just the right viscosity, and since the viscosity varies with the temperature, we must have the right viscosity at the working temperature of the oil film. Some engines run hotter than others and need a heavy oil. Others run cool and a viscous oil could not feed fast enough to give proper lubrication.

An oil with a low surface tension with respect to the metals of which the cylinder and piston are composed, will adhere more tightly to them than an oil with a higher surface tension; in other words, its adhesiveness will be greater. A high adhesive quality will also prevent the oil from being blown off the cylinder wall and out of the clearance space and therefore will improve the piston seal. Unfortunately, it is a difficult matter at present to measure the adhesion of oils with respect to different metals, and we have to depend entirely upon actual running tests to determine the value of the various lubricants.—"Power."

UTILIZATION OF OIL-ENGINE EXHAUST GASES.

By J. L. Chaloner.

From time to time some new type of heat engine invites attention. Claims are made, the scantiest examination of which makes it apparent that deductions have been drawn and conclusions have been arrived at without the slightest consideration having been given to even the most elementary principles of thermo-dynamics. In fact, if what the inventor asserts could be substantiated by practical demonstration, existing methods of heat generation, whether external or internal, are eclipsed continuously and consistently to the extent of anything up to 100 per cent.

With the present state of scientific advancement, one is hardly prepared to accept new principles of heat generation, which must perforce embody revolutionary theorems in order to achieve what they set out to do. There appears to be the impression in some quarters that thermo-dynamical principles may be ignored or overriden provided that the new heat generator or prime mover is blessed with a liberal amount of mechanical complications.

From the very inception of the internal-combustion engine its history has proved that simplicity is the keynote to success, an axiom which inventors would be wise to recognize. In some cases which have come before our

notice it was quite obvious that the engines had at the time of the claim not even got past the "drawing-board" stage. Whether lack of finance or absence of courage was the reason one cannot say. The fact remains, however, that their inventors are still alive, which makes it apparent—at any rate to us—that no actual engine was ever tested in their presence. Carnot's cycle, although not one to be applied in a practical design, indicates the limit of maximum efficiency of any heat cycle with the following relations:

$$\eta \text{ max} = \frac{Q_1 - Q_2}{Q_1}$$

where

Q_1 is the total heat received during the cycle,

Q_2 is the total heat rejected during the same cycle,

and

η is the efficiency of the cycle.

We have got within 70 per cent (on an indicated thermal efficiency base) of this maximum useful heat exchange with the mixed cycle, in which combustion takes place partially at constant volume, partially at constant pressure, and which emyloys a high compression ratio. It has been proved also that by making certain corrections, chiefly concerned with specific heats at high temperatures, that we are today actually within 20 per cent of the ideal —that is, the maximum possible—efficiency. In accordance with the laws of thermo-dynamics, as they present themselves to us at present, we have, therefore, only very limited scope for improving the efficiency of the modern oil engine. The oil turbine of today forms practical proof of this argument.

Examining for a moment the heat balance sheet of an oil engine, it will be seen that the greater portion of the available heat relative to the heat usefully employed is carried away by the cooling water and the exhaust gases. Hence, if there exist an avenue which might be exploited to advantage it should be in the direction of the possible utilization of the heat carried to waste by the circulation water and exhaust gases.

It is not our intention to give the impression that with the above reasoning we attempt to propound a new theory or introduce some startling innovation. We would state that this loss of heat energy has had the attention of all interested parties since the very beginning of the internal combustion era. As a matter of fact, in the days of the gas engine there was even more scope owing to the lower efficiency of the Otto cycle. The difference in a broad classification of the heat energy and its distribution may be seen from Table No. 1:

TABLE NO. 1.

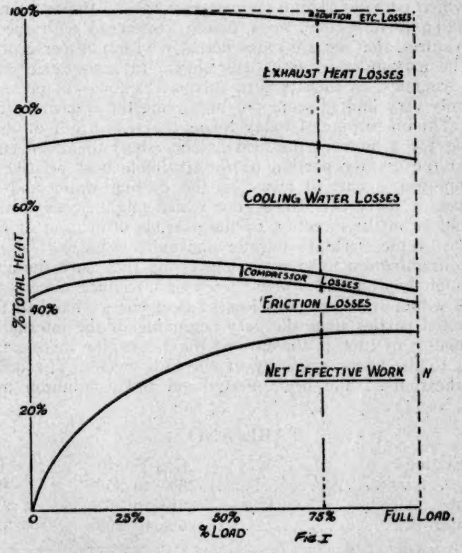
Type of Engine	Gas-Engine	Oil-Engine
Useful heat	36% to 35%	42% to 46%
Exhaust heat	39% to 36%	25% to 24%
Cooling Water	25% to 29%	33% to 30%
Total	100%	100%

No doubt many an engineer has given the utilization of this waste heat his consideration, and we suggest that a useful purpose will be served by a discourse on this important subject. A presentation of pertinent technical data is bound to be interesting and instructive, even if only from a general knowledge point of view. It is hoped that the following discussion will act as a guide—or as a brake, as the case may be—to those sea-going engineers, who in their spare moments would like to calculate for themselves how much bunker they carry for which they get no effective return.

THE HEAT BALANCE SHEET OF AN OIL ENGINE.

For the purpose of establishing a fairly wide base for our proposed investigation, and furthermore to prove some of the conclusions which we hope to establish, we have prepared in tables Nos. II, III and IV a number of heat balance sheets, which cover practically every type of engine which it may be deemed advisable to examine from the point of view of wasted heat utilization.

With the object of simplifying matters, the speed has been taken as the classification factor, and thus it will be seen that Table No. II refers to engines running at 350 R.P.M. and over, Table No. III deals with engines working with a speed between 300 and 150 R.P.M., while finally in Table No. IV engines of 125 revolutions and lower are dealt with. In each case the units are arranged in order of horsepower rating per cylinder, which arrangement would call for the assumption—quite a reasonable one to make—that all cylinders are working under similar conditions as far as each unit is concerned. In Fig. No. I is shown what might be considered



Average heat-balance sheet showing distribution of losses for various loads expressed as percentage of full-load

an average heat balance sheet, and the graph has been arranged in such a way as to indicate the distribution of the losses for various loads, expressed as a percentage of full load.

TABLE NO. II.

HIGH SPEED ENGINES

Unit No.	1	2	3	4
Type and Size	5 B. H. P. 4-cycle	B. H. P. 4-cycle	B. H. P. 4-cycle	B. H. P. 4-cycle
Cylinder Dimension (in M/m.)	116x150	250x350	330x400	250x490
Speed (Revs. per min.)	600	400	405	400
B. H. P. (per Cylinder)	4.76	39.6	74	74
M. E. P. (Lbs. per sq. in.) B. H. P.	63.5	73.7	68.7	70.0
(Lbs. per sq. in.) I. H. P.	109.0	114.0	99.2	104.5
(Lbs. per sq. in.) F. H. P.	25.6	27.8	31.5	31.5
Heat Balance: Losses: Mech. Friction	.....	.....	.....	.....
Heat Balance: Losses: Compressor	18.2	15.2	12.8	15.5
Heat Balance: Cooling	26.8	28.6	36.6	34.3
Heat Balance: Exhaust	29.4	28.4	18.6	18.7
Heat Balance: Total %	100.0	100.0	100.0	100.0
Compression: Pressure (lbs. per sq. in.)	461	.....	570	520
Compression: Ratio	.....	.....	15.7	15.6
Air Excess Co-efficient	.....	.....	.....	.....
Exhaust Temperature (°Fahr.)	686	....	.....	.....

TABLE NO. III.

Engine Unit No.	1	2	3	4	5	6	7	8	9
Size and type	8 H. P. 4-cycle	12 H. P. 4-cycle	15 H. P. 4-cycle	30 H. P. 4-cycle	30 H. P. 4-cycle	45 H. P. 4-cycle	30 H. P. 4-cycle	60 H. P. 4-cycle	120 H. P. 4-cycle
	165x	185x	215x	255x	260.1x	290x	310x	375x	380x
Cylinder Dimension (in M/m.)	269.6	320.7	340	388	400.3	452	463	550	560
Speed (Revs. per minute)	270.3	277	235	229	179.8	179	245	166	180.1
B.H.P. (per Cylinder)	8.62	12.14	15.56	25.3	30.8	38	47.8	60.3	63.2
M. E. P. (lbs. per sq. in.) B.H.P. (Base)	73.0	65.3	68.6	72.7	67.7	73.8	77.0	77.0	70.0
(lbs. per sq. in.) I.H.P. (Base)	94.4	87.0	96.0	103.0	94.0	104.0	106.0	103.0	91.0
Heat Balance: Effect. Horse-Power	27.6	36.0	28.9	32.4	27.2	29.3	30.5	33.7	33.8
Heat Balance: Losses: Mech. Friction	6.7					9.3	7.5		6.0
Heat Balance: Losses: Compressorn	2.6	9.3	8.6	11.3	13.5	10.4	2.8 12.1	4.0 11.5	11.3	2.9 9.8
Heat Balance: Cooling water	18.2	34.9	30.7	30.3	21.9	26.0	25.4	28.0	24.8
Heat Balance: Exhaust Gases	44.9	30.5	29.1	23.8	40.5	32.6	32.6	27.0	31.6
Heat Balance: Total percentage	100	100	100	100	100	100	100	100	100
Compression: Pressure (lbs. per sq. in.)	490	425	495	495	.....	485	490	.....	460
Compression: Ratio	.....	14.36	11 58	.....	.....	13	15.3	.....	15.4
Air Excess Co-efficient	.....	1.93	1.90	.....	1.92	1.8	1.45	.....	.....
Exhaust Temp. (°Fahr.)	.....	633	780	.....	722	710	535	.....	.....

TABLE NO. IV.

Unit No.	1	2	3	4	5	6	7	8
Size and Type	1,500 I. H. P. 2-cycle	1,620 I. H. P. 4-cycle	1,600 I. H. P. 4-cycle	2,220 I. H. P. 4-cycle	2,000 I. H. P. 4-cycle	2,140 I. H. P. 4-cycle	1,680 I. H. P. (opposed-piston) 2-cycle	2,240 I. H. P. 4-cycle
Cylinder Dimensions (in ins.)	18" x 37"	24½" x 39"	24½" x 38½"	28½" x 47"	25¾" x 39¾"	26¾" x 47½"	18½" x 25"	30" x 48"
Speed (Revs. per minute)	115	118	124	96.5	123	100	115	105
B.H.P. (per Cylinder)	188	308	311	225	233	268	309	308
M.E.P. (lbs. p. sq. in.) B.H.P. (Base)	60.0	76.0	75.5	71.0	72.0	82.0	77.0	68.5
(lbs. p. sq. in.) I.H.P. (Base)	92.0	99.0	98.0	99.0	99.0	110.0	101.0	82.0
Heat Balance: Effect. Horse-Power	31.0	33.0	33.6	33.0	33.2	33.1	31.4	32.9
Heat Balance: Losses: Mech. Friction	.....	.....	.....	.....	.....	.....	.....	.....
Heat Balance: Losses: Compressor	10.3	10.0	10.2	13.4	12.4	11.3	9.9	6.8
Heat Balance: Cooling Water	.....	.....	.....	.....	.....	.....	.....	.....
Heat Balance: Exhaust Gases	58.7	57.0	56.2	53.6	54.4	55.6	58.7	60.3
Heat Balance: Total %	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0
Compression: Press. lbs. p. sq. in.)	490	.....	500	495	510	500	495	.....
Compression: Ratio	.....	.....	.....	.....	.....	.....	.....	.....
Air Excess Co-efficient	.....	.....	.....	.....	1.58	.....	.....	.....
Exhaust Temperature (°Fahr.)	.....	600	550	653	.....	.....	.....	.....

INES

	4	5	6	7
	450	450	1,650	1,700
P.	B. H. P.	B. H. P.	B. H. P.	B. H. P.
	4-cycle	4-cycle	2-cycle	4-cycle
400	320x420	315x430	430x500	430x530
	400	400	350	370
	74	74	275	281
.7	70.0	69.3	66.0	94.0
.2	104.5	99.0	100.5	118.0
.8	31.5	31.3	26.8	29.9
	.....	8.7	.....	8.6
.6	15.8	4.5 13.2	16.4	3.5 12.1
.8	34.3	37.3	26.6	36.3
.6	18.7	18.2	30.2	31.7
.0	100.0	100.0	100.0	100.0
	520	.....	490	525
.7	15.6	.....	.....	15.6
..	.....	.....	1.60	1.43
..	.....	.....	885	930

8	9	10	11	12	13	14	15	16
				200	300	1000	800	1000 H. P.
H. P.	120 H. P.	200 H. P.	70 H. P.	H. P.	H. P.	H. P.	H. P.	(double
ycle	4-cycle	4-cycle	4-cycle	4-cycle	4-cycle	4-cycle	4-cycle	acting) 4-cycle
75x	380x	370x	400.5x	430x	510x	550x	660x	740x
50	560	560	600.5	670	610	800	840	1060
60	180.1	188.2	158.8	160.5	200	151	145	124.9
60.3	63.2	66.5	60.63	96.9	123.8	167.8	196.5	501.0
77.0	70.0	76.0	73.8	81.0	64.5	76.0	60.0	66.9
63.0	91.0	94.0	96.0	106.5	85.0	98.0	78.6	85.0
33.7	33.8	33.6	31.9	34.2	34.2	35.6	34.8	29.5
	6.9	8.8	7.5					
11.3	2.9 9.8	2.3 11.1	2.1 9.6	10.9	10.8	10.2	10.4	8.2
28.0	24.8	27.2	27.6	26.5	26.3	26.6	25.7	26.6
37.0	31.6	28.1	30.9	28.4	28.7	28.6	29.1	25.7
00	100	100	100	100	100	100	100	100
.....	460	.....	490	503	483	500	505	465
.....	15.4	.....	15.4	.....	15.3	.....	15.4	.....
.....	.....	.....	.....	.....	.....	.....	.....	.....
.....	.....	.....	645	752	.....	680	.....	563

	8	9	10	11	12	13	14
			1,400				3,000
-	2,340	1,700	I. H. P.	3,200	2,000	2,900	I. H. P.
	I. H. P.	I. H. P.	(Still)	I. H. P.	I. H. P.	I. H. P.	(opposed-
	4-cycle	2-cycle	2-cycle	4-cycle	2-cycle	2-cycle	piston) 2-cycle
	30"x	23½"x	22"x	26½"x	26¼"x	25½"x	22½"x
	48"	37"	26"	48½"	26½"	48"	45½"
	105	102.2	125	115	103.4	100	76.2
	208	316	327	328	402	480	535
)	68.5	74.0	75.8	75.0	75.0	73.0	88.2
)	83.0	102.2	91.4	91.5	92.5	105.0	109.4
			(29.9)				
	32.9	33.0	31.0	33.3	35.4	31.4	34.5
	.....	.....	.....	.....	.....	.....	.....
			(9.1)				
)	6.8	12.6	11.0	7.3	8.4	13.8	8.0
			Condenser(40)				
	.....	.....	20.0	.....	.....	.....	.....
			(11)				
	60.3	54.4	38.0	59.4	56.2	54.5	57.5
	100.0	100.0	100.0	100.0	100.0	100.0	100.0
	.....	.....	300	.....	420	.....	300
	.....	.....	.....	.....	.....	.....	.....
	.....	.....	.....	.....	.....	.....	.....
	.....	404	675	.....	600	.....	570

As will be seen, we are mainly concerned with the heat passing away in the exhaust gases and the cooling water, and as the former presents rather more difficulties in arriving at some practical conclusion, we propose to deal with that part of the heat loss first.

EXHAUST GASES AND THEIR HEAT VALUE.

For the purpose of calculating the potential heat of any substance we take the product of the weight of the substance, its specific heat and temperature. In order, therefore, to find the available heat in the exhaust gases we must know their weight or volume, we must have an analysis in order to determine the specific heat of the products of combustion, and finally we must have the temperature of the gases on leaving the engine cylinder.

EXHAUST GAS TEMPERATURES.

The temperatures of the exhaust gases are quoted in several cases on Tables No. II, III and IV. These figures only act as a general guide of temperatures to be expected, as on closer examination there is apparent inconsistency between the temperatures for the different types. It must be remembered that the results were obtained by a number of observers, who used different measuring devices in different positions relative to the exhaust valve or port.

TABLE NO. V.

Unit No: (see Table No. IV.)	Exhaust Temperatures °F. @ % Full Load. 110%	100%	75%	50%	25%	Type
3.....	594	540	470	401	342	4-stroke
9.....	584	459	368	302	240	2-stroke
10.....	756	678	579	494	...	2-stroke
12.....	635	590	500	397	284	2-stroke

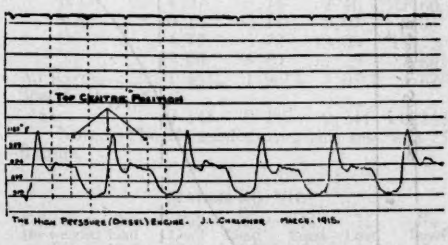


FIG. 2.—TEMPERATURE VARIATIONS OVER SIX COMPLETE CYCLES AT FULL POWER.

Again the shape of the combustion chamber, the atomizing device, the degree of combustion, the grade of fuel are only a few of the many factors which determine the actual temperature of the gases on leaving the cylinder.

Load conditions, general design and test methods are obviously not uniform and thus affect the results to a definite degree. For instance, the heat transference varies with the speed; it is obviously greater with slower speed. Again the ratio of cooled surface to cylinder volume controls the relative amount of heat passing into the gas and cooling water, more heat

being absorbed by the water with an increasing ratio of surface to volume. The speed of the engine has a specific influence on the exhaust temperature, the latter varying inversely as the speed.

Available heat is somewhat higher for the two-cycle engine than for the four-cycle, as might be anticipated. The tables, however, do not bring out this point as positively as one might desire, the net difference being only about 1 per cent. The reason is most probably found in the influence which the scavenging air, both with regard to volume and pressure, exercises over the exhaust.

Slight variation in heat distribution for small and large cylinders is interesting and bears out the general experience that in the oil engine a relatively large increase in cylinder dimensions does not mean a decrease of fuel consumption per unit of similar dimensions.

Mean-effective pressure has a definite relation to the exhaust temperature not only for two engines running at the same speed, but also for the same engine at reduced power. In Table No. V a number of test data are quoted for various loads expressed as a percentage of normal rating.

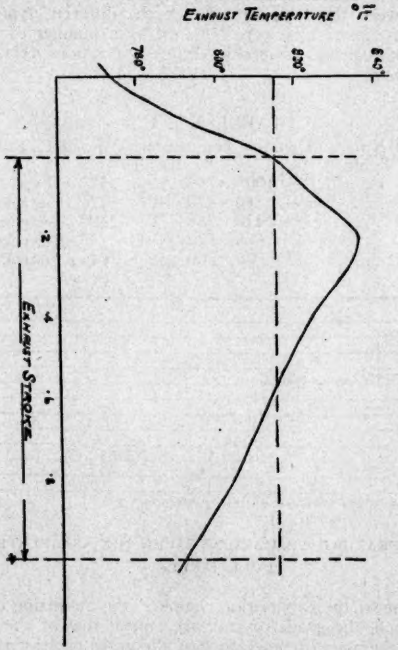


Fig. 3.—Curve of Temperature Readings Throughout Exhaust Stroke.

These temperatures for the four engine units can only be compared with each other by making such corrections as become necessary in view of what has been said previously with regard to the influence of various factors. An

examination of the particulars on Table No. IV with specific reference to the four units mentioned in Table V will be an advantage:

Some actual measurements by Woolf are very interesting and instructive in this connection. It must be remembered, however, that the engine was only rated at 40 B.H.P. (12.5 inches diameter by 19 inches stroke), the speed being 190 R.P.M. The temperatures were obtained by direct measurement; a pyrometer was fitted into the combustion chamber on the exhaust valve side. The recording terminal was ⅛ inch long and about ⅛ inch distant from the cylinder wall. It was secured in a position parallel to and 1⅛ inch from the under side of the cylinder cover. Continuous temperature records were taken under different load conditions, and in Fig. 2 we reproduce a chart showing the temperature variations over six complete cycles with the engine developing full power. Fig. 3 shows an enlarged view of the exhaust stroke, and indicates the gradual and comparatively small drop in the temperature throughout the exhaust stroke.

SPECIFIC HEAT OF EXHAUST GASES.

The fuel being a hydro-carbon, one may expect the composition of the gases to include CO_2, O and N as the main constituents, with SO_2, H_2O and possibly CO in small quantities. Here again we resort to the use of two tables to indicate the conditions as they exist under various loads. Speed, load and atmospheric conditions on one hand, grade of fuel and air excess co-efficient on the other are responsible for the percentages of the various constituents.

TABLE NO. VI.

Composition of gas (by weight)	Full load	¾ Load	½ Load	¼ Load
CO_2	15.10	11.28	7.35	4.85
CO	0.00	0.00	0.00	0.01
H_2O	4.97	3.73	2.40	1.60
SO_2	0.16	0.11	0.07	0.03
O_2	4.77	8.78	13.03	15.72
N_2	75.00	76.10	77.25	77.80
Air Excess	1.45%	1.96%	3.05%	4.67%
Specific Heat of gas	0.248	0.247	0.244	0.242

Note:—The oil used during this test was 25° Baume.

TABLE NO. VII.

Exhaust gas (by weight)	Over-load	Full Load	¾ Load	½ Load	¼ Load	No Load
CO_2	13.9	11.2	9.1	6.8	5.4	3.4
CO						
H_2O	5.6	4.6	3.9	3.1	2.8	1.6
O_2	7.6	11.2	13.0	15.5	18.6	19.3
N_2	73.0	73.0	74.0	74.6	75.0	75.7
Air Excess	1.53%	1.9%	2.35%	3.2%	3.95%	6.5%
Specific heat of gas	0.260	0.256	0.253	0.249	0.247	0.243

Note:—The oil used during these tests was 29° Baume.

Direct air measurements and calculations of the excess air co-efficients from the exhaust-gas analysis agree so closely together as to make the latter method sufficiently accurate for all practical purposes. This is an important concession in view of the simple and familiar way, in which exhaust gas tests are carried out. One must remember that only four-cycle

engines can be dealt with in this simple manner, and that in the case of the two-cycle engine careful calculations of the scavenging-air quantities must be made. In Tables No. VI and VII two sets of exhaust-gas analyses are quoted for various parts of the full load output of the engine.

Attention is drawn to the fact that the above analyses are determined on a gravi-metric scale. In passing it may be useful to recall that air consists of 79.16 per cent of nitrogen and 20.84 per cent of oxygen by volume, but 76.85 per cent of nitrogen and 23.15 per cent of oxygen by weight. Whilst it is rather difficult to give any typical illustration of gas analyses for two-cycle and four-cycle engines, Table No. VIII has been prepared with a

TABLE NO. VIII.

Exhaust gas Analysis

Volumetric % of Constituents	4-stroke marine engine	2-stroke marine engine	Specific Heat (constant pressure)	Specific weight lb. p. cbft.
CO_2	9.0	5.0	0.240	0.0385
CO.....	0.0	0.1	0.251	0.0245
H_2O	2.0	2.0	0.480	0.0160
SO_2	0.1	0.1	0.260	0.0560
O_2	7.0	12.0	0.230	0.0280
N_2	81.9	80.8	0.247	0.0246
Air(forcomparison).....	....	....	0.238	0.0253
Mean specific heat of gas..	0.2498	0.2497	....	.....
Mean specific weight of gas	0.0281	0.0276	....	.....

view to indicate what the results might be expected to be, provided normal conditions are retained. The figures refer to Engine Units Nos. 5 and 12 on Table No. IV, and the comparison is very instructive and interesting.

Values for specific test are added, and we have assumed a temperature of 1,200 degrees Fahrenheit as the exhaust gases leave the combustion chamber. The specific heats and specific weights are the values for this temperature.

WEIGHT OF EXHAUST GASES.

During the chemical process of combustion the carbon and hydrogen in the oil combine with the oxygen in the air. As in the case of coal, therefore, a definite amount of air must be provided to produce complete combustion.

Owing to the conditions inside the combustion chamber, it is found that the amount of air, calculated in the usual way from the analysis of the oil, is not sufficient to produce the desired degree of combustion. As in the case of coal, a certain additional volume must be provided for over and above the theoretical amount. As has been noted, this amount is not as large as is found necessary with coal. The calculations for air requirements are quite simple and no reference is deemed necessary at this stage.

One may like to compare one's coal results with the modified conditions when using oil, and here one must remember several important differences between oil and coal.

The hydrogen contents are 5 per cent for coal, but 12 per cent for oil.

The maximum possible percentage of carbonic acid varies as the hydrogen carbon ratio, and in this lies the explanation of the comparatively low percentage of CO_2 as indicated in Tables VI, VII and VIII. Take a pound

of coal, dried and ash-free, having 80 per cent carbon and 5 per cent hydrogen. The maximum percentage of CO_2 possible is, say, 18¼ per cent.

A pound of oil, free from water and mechanical impurities, has 84 per cent carbon and 12 per cent hydrogen.

The highest possible yield of CO_2 would not exceed 15¾ per cent.

For the oils which are used for marine oil engines the hydrogen content varies from 11 to 12 per cent; the corresponding carbon is 82 to 84 per cent.

It will thus be seen that the theoretical amount of air is between 13 and 14 pounds, or 160 to 175 cubic feet (free air at 60 degrees Fahrenheit).

Calculations will reveal that for an oil of 34 degrees Baumé about 14 pounds of air are required, and that a 24-degree Baumé grade requires about 13 pounds. As the gravity increases, so the amount of air becomes less. As to the excess air we find from a large number of tests with different engines that 80 and 100 per cent for four-cycle and two-cycle types respectively gives the best all-round results. Expressed as excess air co-efficients increase in proportion to the values quoted in Tables Nos. VI and VII. For the purpose of an approximation of the weight or volume of exhaust gases we suggest the following:

$$\text{H. P.}_B = \frac{\text{P. L. A. N.}}{33{,}000} = \frac{P_B}{3.8}\ \frac{\text{(L. A. N.)}}{(144\ \ 60)}$$

$$= \frac{P_B}{3.8} \times V_A$$

Where V_A = volume of air per sec. (if volumetric efficiency is 100 per cent)

$$\therefore V_A = \frac{\text{H. P.}_B \times 3.8}{P_B}$$

Where H. P.$_B$ = B. H. P.

P_B = 65 for 2-stroke
70 for 4-stroke

$\therefore V_A = 0.058$ H. P. cu. ft. per sec. (2-cycle)

$V_A = 0.054$ H. P. cu. ft. per sec. (4-cycle)

We may say that: —

$$2\ V_A = V_E$$

hence

$V_E = 0.116$ H. P. cu. ft. per sec. (2-cycle)

$V_E = 0.108$ H. P. cu. ft. per sec. (4-cycle)

where V_E = volume of exhaust gases.

To illustrate the above, we will quote the following example: —

B. H. P. . . . . = 3,000

Fuel Consumption . . . . . = 0.42 lb. per B. H. P. hour.

Theoretical air per lb. of fuel . . . = 14 lbs.

Air excess co-efficient . . . . . = 1.8

Mean specific weight of exhaust gas (see Table No. VIII) . . . . . = 0.281

$$\therefore \text{Volume of Exhaust per sec.} = \frac{3{,}000 \times .42 + 3{,}000 \times 0.42 \times 25.2}{60 \times 60 \times 0.281}$$

$$= 326 \text{ cu. ft. per sec.}$$

Using above formula Volume of Exhaust Gas . . . . . = 0.108 × 3,000
= 324 cu. ft. per sec.

It will be seen that results are very close.

In using the above approximate formula one must remember, however, that it is on the high side on account of the volumetric efficiency being taken at 100 per cent. As actually the volume drawn into the cylinder at each stroke is less than the capacity of the cylinder, a correction may be made with advantage. It will be found that the volumetric efficiency is about 90 per cent.

HEAT AVAILABLE IN EXHAUST GASES.

The data as enumerated in the previous statements indicate the potential heat energy, and the question arises as to how much heat can be usefully employed. The radiation losses are, of course, considerable, although the actual percentage depends on the method and extent of utilizing the available heat.

The drop in pressure, the frictional losses and the ineffective heating surface are responsible for a loss of heat, which is unavoidable. In a certain installation exhaust temperatures were measured at various points, as recorded in Table No. IX. The exhaust pipe was water-cooled and in the

TABLE NO. IX.

Distance from Exhaust Valve	Diameter of Exhaust Pipe	Temperature °Fahr.
9″	6″	685
6′0″	8″	668
26′0″	8″	480
33′0″	8″	356

length under observation there was only one bend. The outlet cooling water temperature was about 120 degrees F.

Taking the average figures for the factors which make up the total potential heat in exhaust gases, we can now establish the total heat in exhaust gases per horsepower as follows:

Total heat = 0.108 × 0.028 × 0.250 × 1200
= 0.91 H.P. B.T.U. per sec.
= 3,275 H.P. B.T.U. per hr.

Allowing for volumetric efficiency (as we have assumed 100 per cent in our approximation) radiation and a reasonable temperature of the exhaust gases on leaving the waste heat boiler, we find that:

Total available heat = 1,650 B.T.U. per hour per horsepower.

If we assume the exhaust gases to be utilized for steam raising purposes, then with an exhaust boiler efficiency of 60 per cent we get 990 B.T.U. available heat. The boiler pressure may conveniently be fixed at 100 pounds, hence

Steam available = 0.85 pound per H.P. hour or
1.0 pound per H.P. hour from and at 212 degrees F. With a 3,000 H.P. installation therefore we may expect an output of 90 to 135 H.P. for auxiliary purposes whilst at sea (30 to 25 pounds of steam per I.H.P. hour) steering gear, circulating-pumps, ballast-pump, oil transfer pump, feed donkey and dynamo can all be steam-driven sets.

In port the cargo winches will necessitate the exhaust-boiler being arranged for oil-firing, and in all probability would demand a larger evaporation than when at sea. Hence in most cases the boiler rating will have to be designed for port conditions and at sea the boiler efficiency would be somewhat lower than 60 per cent, as suggested above.

AVAILABLE HEAT IN JACKET-COOLING WATER.

In the above calculations no account has been taken of the available heat in the cooling water. If the cooling water is taken from the sea and pumped overboard again, then sufficient fresh water must be carried for the exhaust-boiler together with the necessary cooling plant and exhaust returns from the various auxiliaries as far as possible, unless an evaporator is fitted.

If fresh water is used for cooling water, then the boiler-feed can be taken direct from the engine jackets. There is an additional advantage when using fresh water for jacket cooling, because the circulating water outlet temperature can be raised to 150 degrees to 170 degrees F. as against 110 degrees F. in the case of sea water. There would thus be an additional gain of useful heat with the correspondingly larger amount of steam per H.P. hour.

As is well known by now, the Still combination steam and internal-combustion engine uses this system to great advantage with a resultant gain to the thermal efficiency of the main engines. Some 18 per cent of useful heat is recovered, raising the thermal efficiency of the Still engine from 33 per cent to 39 per cent.

It is hoped that in the foregoing certain data have been submitted which may prove interesting to those engineers who have considered the question of utilizing some of the heat which at present is lost for all time. As in most problems of this nature, one cannot be dogmatic, but must confine oneself to recording the various aspects of the problem in a general way.

The question of total net effect depends on the extent to which the combined heat of the circulating water and exhaust gases can be utilized. It will be found that under certain conditions only the exhaust heat may be used to advantage. There again one must investigate the percentage of heat which will be extracted from the exhaust gases by the cooling water before the gases reach the boilers.

It is hoped that in a future reference to this subject, some actual designs of exhaust boilers will be discussed and to what extent they have justified their installation on board of motorships.—"Motorship," January, 1923.

A NEW DOUBLE-ACTING SEMI-DIESEL ENGINE.

THE REED TWO-CYCLE TYPE. A REVOLUTIONARY TWO-CYLINDER 50 H.P. OPEN DESIGN, USING AIR INJECTION.

While a good deal of attention has been given to the two-cycle double-acting Diesel engine, it should not be forgotten that hot-bulb machinery having similar characteristics has been in course of development for some time past. The following description relates to the first two-cylinder 50 H.P. engine completed under the Reed patents. This plant, of which various photographs and plans are reproduced, is being made by the Reed Patent Engine Co., Hythe, Southampton, and is in many ways one of the most remarkable engines yet constructed.

As the plant works on the double-acting two-cycle principle, hot bulbs are necessarily fitted at the top and bottom of the main cylinders. Referring to the sectional end elevation, it will at once be seen that a most unusual arrangement of mechanism is instituted. Through the central main working piston passes a rigid connecting pin, to which are secured scavenging plungers, disposed athwartships. Of the open type, the engine has no piston rod in the generally accepted sense of the word, but includes a couple of side rods, which derive their motion from the horizontal connecting pin, and are coupled at the lower end to a specially shaped cross-head guide. A point which is possibly of minor importance in relation to other features is that the engine is off-set.

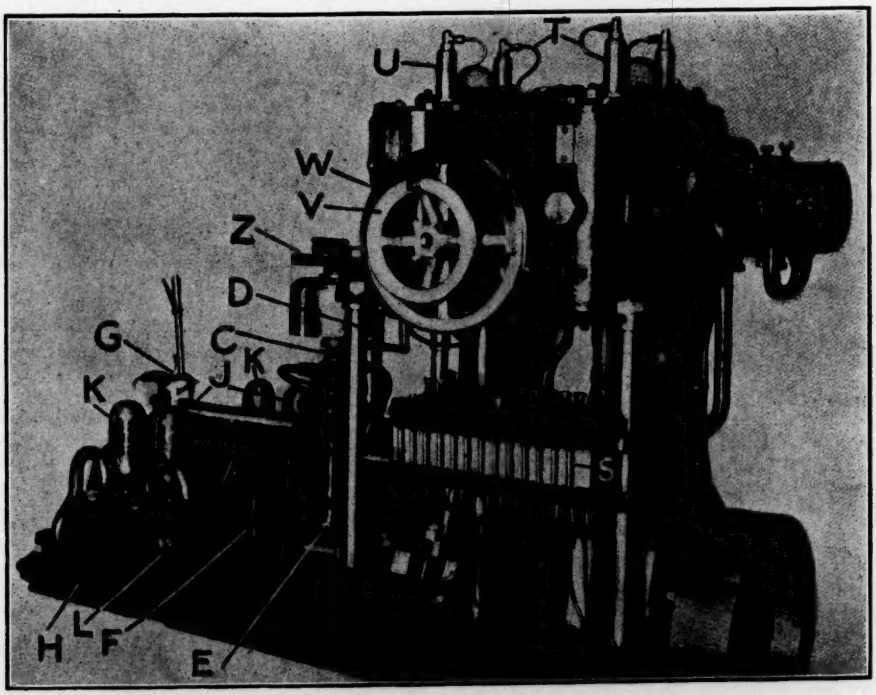
According to the design illustrated, the two vertical side rods attached to the cross-head are provided with spring-loaded packing pieces. The main piston partially uncovers the slot provided in the cylinder walls for permitting motion of the connecting pin, but it should be remarked that an alternative method of construction may be employed, incorporating a piston longer than that shown; with the modified arrangement, the slots are not uncovered, and the side-rod spring-loaded pads are not used. The combined area of the scavenging pump plungers is 25 per cent greater than that of the main pistons, while the air compressed by the plungers is discharged to the cylinders at a pressure of approximately 15 pounds per square inch above atmospheric. The upper combustion chamber is scavenged from the lower ends of the pumps, and vice versa, the products of combustion exhausting through a centrally disposed port into a manifold at the back of the engine. In order to appreciate various details in connection with the equipment, it will be necessary to refer to the lettered illustrations. In each case, the letters employed refer to the same, or duplicated, portions of the mechanism. The bore of the cylinders is 6 inches and the stroke 6 inches, the speed being 300 R.P.M.

MECHANICAL DETAILS.

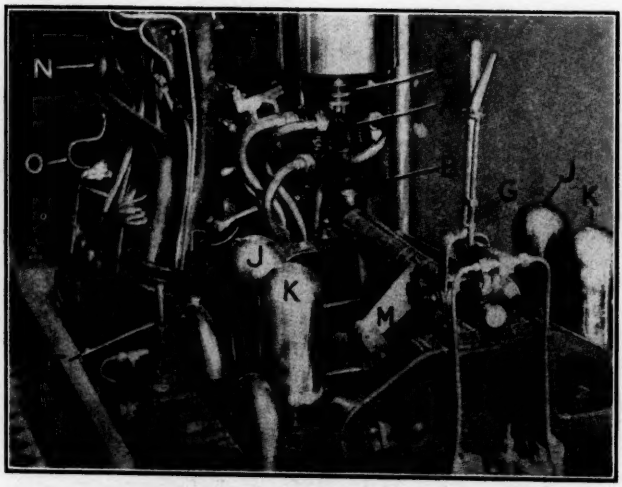
Compressed air is used for starting purposes, the distributor (A) being driven from a vertical shaft operated from the main crankshaft, and having leads to the top and bottom ends of each cylinder. The injection of the fuel is also accomplished by means of compressed air, below the starting air distributor being an additional arrangement (B) in connection with four injection air pipes. For the purpose of regulating the governor, hand adjustment is allowed; this device will be seen at C in several illustrations, directly below the governor casing.

Two eccentrics and rods (D) work the fuel pumps, the drive being forward of the bevel gearing (E), which works the vertical shaft to which reference has been made. In duplicate, the circulating and bilge pumps involve some interesting mechanism; for example, a pair of whistles is provided, one of these being shown at F. Normally, opposite pumps are in operation; the bilge pumps are out of action unless the circulating water fails, when the whistle gives warning to the driver and the bilge pumps automatically take over the duty of maintaining the water supply to the jackets and pistons. The bilge pump clutch (G) drops into operation without attention on the part of the driver.

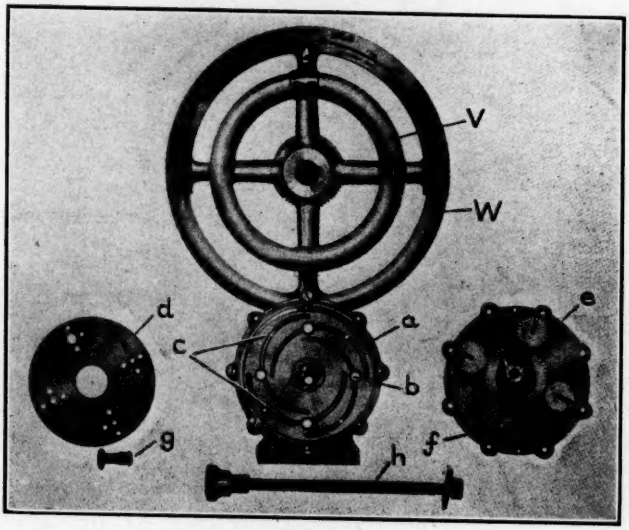
For changing over the pumps, a horizontally arranged transformer (H) is fitted. Air vessels (J) will be observed above the circulating pumps, and a similar arrangement (K) is provided in the case of the bilge pumps. Forward of these pumps are the air compressors, two in number, and arranged horizontally on opposite sides of the main shaft. The starboard machine will be observed at L. An eccentric on the main shaft is coupled to



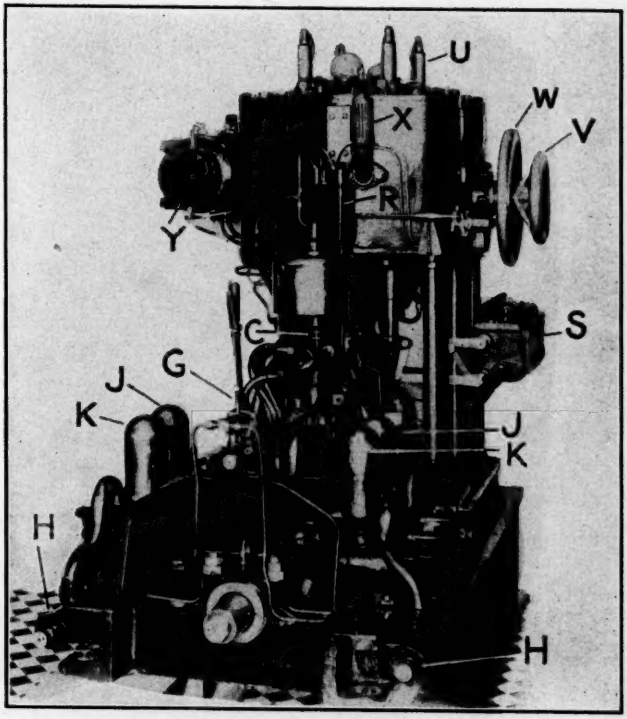
CONTROL SIDE OF THE REED DOUBLE-ACTING TWO-CYCLE ENGINE.



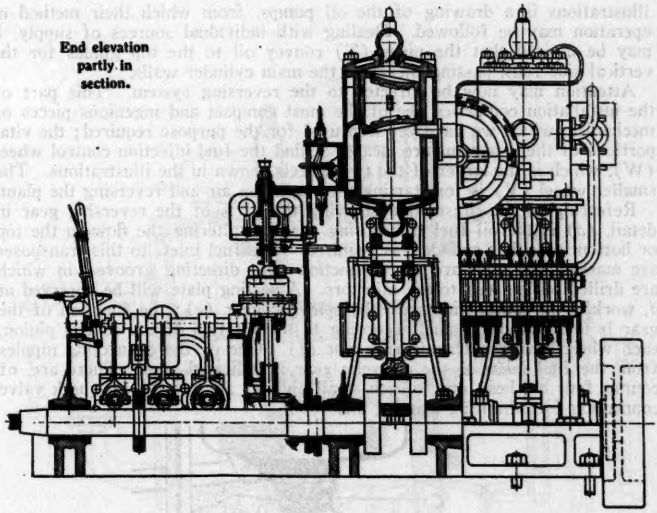
After-End, Showing Pumps.



Control Gear.



General View, Looking Forward.


End elevation partly in section.

the rocking lever (M); this in turn is attached to an overhead horizontal shaft provided with levers for reciprocating the bilge and circulating pumps together with the air compressors.

Each piston is cooled by water passing up one of the vertical side rods, connection being established with an inclined hole in the connecting pin. The return water is conveyed down the other rod, finally passing through a rocking pipe to the discharge. The main circulating pipe to the jackets and pistons is at N, while one of the delivery pipes conveys water to the forward piston, as indicated at O, the inlet rocking pipe to the after piston being marked P.

LUBRICATING ARRANGEMENTS.

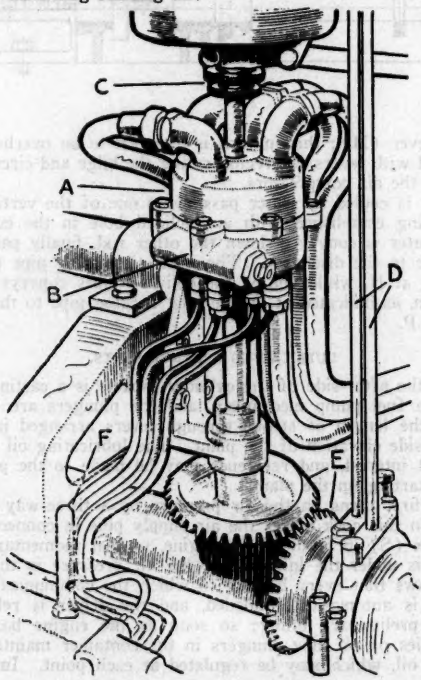
Secured to the after-side of the cylinder jackets is a casting (R) incorporated in the fuel pump mechanism, and the plungers are subject to an alteration in the length of stroke through levers arranged in the manner shown in the side elevation of the plant. The lubricating oil arrangements are of distinct interest, and reference may be made to the procedure followed when starting up the plant.

The driver first turns on the air pressure by a three-way cock. In its neutral position this cock places the air supply pipe in connection with the lubricating box (S) in front of the engine, so that momentarily the oil in the container is under the influence of the air pressure in the system. It therefore follows that every bearing to which the communicating oil pipes are conveyed is automatically flooded, and the driver is relieved of the necessity for preliminary oiling; so soon as the engine has commenced running, a series of oil pipe plungers in the container maintains a supply of lubricating oil, which may be regulated at each point. Included in the

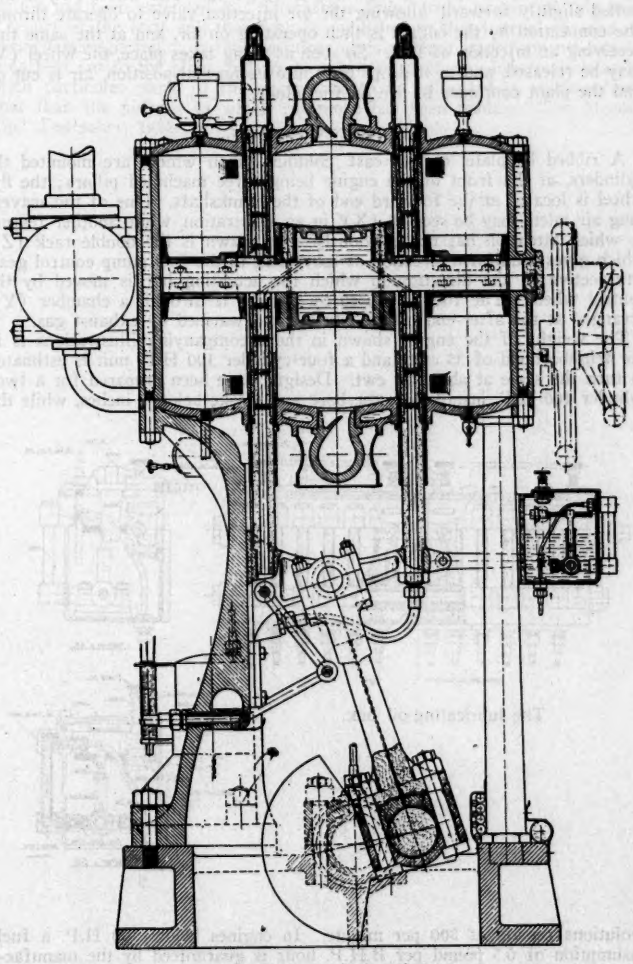
illustrations is a drawing of the oil pumps, from which their method of operation may be followed. Dealing with individual sources of supply, it may be noticed that the pipes (T) convey oil to the top guides for the vertical side rods, passing thence to the main cylinder walls.

Attention may now be directed to the reversing system. This part of the installation comprises one of the most compact and ingenious pieces of mechanism which we have yet seen used for the purpose required; the vital portions of the apparatus are located behind the fuel injection control wheel (W), which is the larger of the two wheels shown in the illustrations. The smaller wheel (V) is for starting the engine on air and reversing the plant.

Referring to the illustration showing the parts of the reversing gear in detail, (*a*) is the oil fuel transposing chamber altering the flow to the top or bottom of either cylinder, as required. The fuel inlets to this transposer are marked (*b*), and are in conjunction with directing grooves, in which are drilled outlets (*c*) to the injectors. A sealing plate will be observed at *d*, working in conjunction with the epicyclic gear (*e*); the function of the gear is to transfer the fuel according to the rotation of the central pinion, each wheel having a transferring slot (*f*). One of the connecting nipples from the fuel inlets to the epicyclic gear is indicated at *g*. There are, of course, four of these, one to each small pinion. At *h* is the air check valve connection for admitting starting air.



STARTING AND INJECTION AIR DISTRIBUTOR.



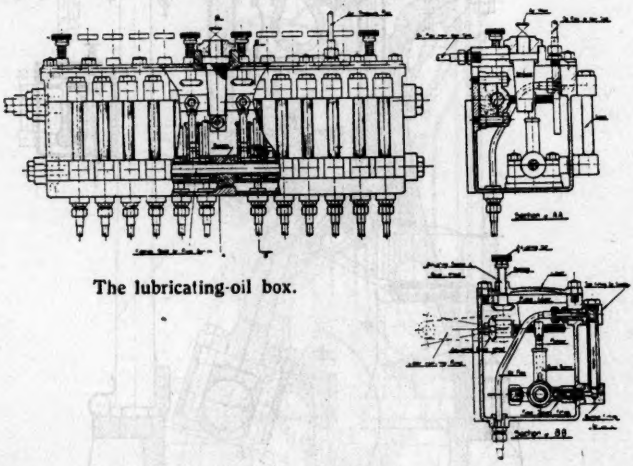
Sectional end elevation.

Assuming that it is desired to start the engine, the three-way air cock to which reference has been made is opened, the center wheel (V) then turned to the air-starting position, and the controlling wheel (W) adjusted as desired, *i.e.*, one-quarter, one-half, or full load. The wheel (V) is then pulled slightly forward, allowing the air injection valve to operate through the connection *h;* the engine is then operating on air, and at the same time receiving an injection of fuel. So soon as firing takes place, the wheel (V) may be released, and as it drops back into its normal position, air is cut off and the plant continues to fire on fuel alone.

FUEL CONSUMPTION.

A ribbed bedplate carries cast columns upon which are mounted the cylinders, at the front of the engine being three machined pillars; the flywheel is located at the forward end of the crankshaft. One of the scavenging air inlets may be seen at (X), in an illustration, while another feature to which attention has not previously been drawn is the double rack (Z), which gives a uniform direction of movement to the fuel pump control gear, irrespective of the direction in which the actuating bar is moved by the control wheel. The fuel is heated by passing it through a chamber (Y), arranged at the after-end of the silencer and warmed by exhaust gas.

The weight of the engine shown in the accompanying illustrations is in the neighborhood of 35 cwt., and a four-cylinder 100 H.P. unit is estimated to turn the scale at about 50 cwt. Designs have been prepared for a two-cylinder 100 H.P. installation, the bore and stroke being 9 inches, while the



The lubricating-oil box.

revolutions remain at 300 per minute. In engines below 100 H.P. a fuel consumption of 0.5 pound per B.H.P. hour is guaranteed by the manufacturers, who claim that the engine will run successfuly on a good quality of Diesel oil, as distinct from gas oil generally employed in semi-Diesel plant. The claim is also made that ordinary Diesel fuel may be used, although no doubt this course is not to be recommended as a general rule.

Up to the time of writing, the trials have been confined to a single-cylinder 25 H.P. unit, which we have not yet had an opportunity of seeing in operation. Arrangements are being made by the builders to carry out a month to six weeks' continuous trial on a 50 H.P. installation, and an invitation has been extended to "The Motor Ship" to be present at this test. A report of the trial will therefore be published in this journal so soon as circumstances permit. In the meantime certain detail improvements are to be carried out to the plant, but we have not received any information as to which particular parts of the engine are to be subjected to modifications other than the pistons, to which reference has been made,—"The Motorship," December, 1922.

OBITUARY.

REAR ADMIRAL JOHN R. EDWARDS, U. S. N.

The long career of Admiral Edwards was noteworthy for his success in many fields. It was marked throughout by high ability, resourcefulness, and untiring energy. These characteristics were manifest to the end. Of his later years, the *Providence Journal* says editorially: "The period of his retirement was one of helpful activity in many civic matters. In his death, this State and his town (Bristol, R. I.) have lost a high-minded and useful citizen."

The Admiral was born at Pottsville, Pa., in 1853, the son of the Rev. Richard and Margaret (Williams) Edwards. He entered the U. S. Naval Academy in 1871, was graduated as a Cadet Engineer in the class of 1874, became a Chief Engineer in 1895, was transferred to the Line of the Navy in 1899 as a Lieutenant Commander, and in 1911 reached the rank of Rear Admiral. His sea service included cruises on the North Atlantic, Pacific, and Asiatic Stations, with special duty on the Presidential yacht *Despatch* and service during the Spanish-American war on the monitor *Puritan*. During the World War, although then retired, he commanded the Naval Unit of Brown University, and after its disbandment was on special duty at Newport, R. I.

Admiral Edwards' shore duty covered much important work. Some of his assignments stand out pre-eminently as those ordinarily given only to officers of unusual ability. Thus, he was for a considerable period Aid for Inspections at the Navy Department, and for two years was President of the Board of

Inspection and Survey for Shore Stations. Again, in 1912, President Taft designated him as Chairman of the United States delegation to the International Radiotelegraphic Conference of London, assembled to regulate international control of radio communication. His duty in this was of the first importance, since there were thirty-one countries represented and the attitude of some of them was decidedly grasping. In such a conflict with astute minds, the Admiral was at his best. Finally, at his retirement and for some time previously, he was Commandant of the Navy Yard at Charleston, S. C., and was in this the first officer in the history of our Navy, trained originally as an engineer, to hold such an important shore command.

From 1888 to 1891, Admiral Edwards was on special duty as Professor of Mechanical Engineering at the University of South Carolina, where he also studied law and in 1891 was admitted to practice in that State. In 1904, he was detailed as Engineer Officer of the Navy Yard at Portsmouth, N. H. In 1908, he became Inspector of Machinery at the Cramp Shipyards, Philadelphia, Pa., and in 1911 was appointed General Inspector of Machinery for all naval vessels building on the Atlantic coast.

From 1895 until the beginning of the Spanish-American war, and from 1900 onward for about four years, the Admiral served at the Bureau of Steam Engineering, Navy Department. His duty there brought him in close touch with Admiral Melville, then Engineer-in-chief, who became his warm friend. Through this relation, Admiral Edwards was appointed Senior Member of the U. S. Naval "Liquid Fuel" Board whose exhaustive report, issued in 1904, has been long regarded as an engineering classic. This Board was composed of exceptionally able officers, its series of tests lasted for nearly two years, and the expenditures by the Navy Department and by business organizations for the building and operation of the experimental boiler plant were very heavy. For the enlistment

of civilian capital in this work, and largely for the original conception of these tests, credit is due Admiral Edwards. Fuel oil, in fact, was with him a life work thereafter, and he became one of the leading American experts in its advocacy and use.

During his first tour of duty at the Bureau, he was also active in the movement to secure positive rank and Line titles for the engineer officers of the Navy. The agitation for these had lasted since the Civil War, and from 1895 onward it became acute. During this latter stage, Admiral Edwards acted for a considerable period as the unofficial agent of his brother officers in this matter. A number of other officers and ex-officers were also serving ably in various ways, but he spent his energy chiefly in arousing public opinion as to the wisdom and inherent justice of the changes desired, not only by the Engineer Corps, but by a considerable body of Line officers as well.

A fluent and skilful writer and always a favorite with newspapermen, he had ready and continuous access to the daily press and to magazines and reviews. He organized a symposium on the condition of naval engineering in the *North American Review*. He appeared as a lecturer on engineering at the Naval War College in a striking and—considering the situation—decidedly daring address on "The Day After the Battle," which was subsequently printed in full in the *New York World*. He enlisted the active support of several large engineering bodies, including the League of Associated Engineers, an organization of 10,000 professional men, formed primarily to promote remedial legislation for engineer officers of the Navy. In such activities as these, he was tireless. In the writer's view, no single line of endeavor did more than that of Admiral Edwards to pave the way for the successful action later of the Amalgamation Board under Theodore Roosevelt, and for the subsequent passage of the Personnel Bill which transferred all engineer officers to the Line of the Navy.

Admiral Edwards was one of the relatively few men who never lose the buoyancy and eager enthusiasms of their youth, although but little contact with him was needed to know that beneath these characteristics lay a keen brain, strong courage, and a forceful aggressiveness in any cause he thought just. He was always frankly outspoken. A loving cup presented to him by the officers of the U. S. Revenue Cutter Service bears this inscription: "As a token of their esteem of one who has no hesitancy in expressing his good opinion of his friends at all times and in all places." With the crowding memories of an unbroken friendship with him of more than forty years, the writer feels it a high privilege to pay this tribute here.

WILLIAM LEDYARD CATHCART.

BOOK REVIEW.

DIESEL ENGINES, BY A. P. CHALKLEY. D. Van Nostrand Company, 8 Warren St., New York. Price $6. 00.

The rapid growth of the Diesel engine has made a book on that subject become out-of-date in a very short time; perhaps for that reason most authors have become discouraged from attempting to keep up with the development. It is therefore a great pleasure to find a book on Diesel engines which is really up-to-date at the present time and it is believed that Mr. Chalkley's latest revision gives just this.

There is included a short treatise on the general subject of heat engines and a comparison is made between the Diesel and other types. The construction of the Diesel engine is thoroughly described; both land and marine types are covered and the whole is copiously illustrated. Diesel fuels, testing, design and operation are also covered, although the last is probably not as comprehensively covered as the marine engineer would desire.

On the whole, any one interested in Diesel engines will find the book very useful and satisfactory.

ASSOCIATION NOTES.

FINANCIAL STATEMENTS.

Following are statements showing the financial transactions of the Society for the year ending December 31, 1922:

PUBLICATION STATEMENT.

This statement shows the returns from the publication of the JOURNAL and the cost thereof:

	Returns.	
Advertisements		$2,782.00
Dues		6,375.00
Sales		517.70
Subscriptions		2,545.16
		$12,219.86
	Costs.	
Printing	$6,716.29	
Engraving	830.68	
Drafting	33.00	
Manuscript	960.00	
Commission	258.66	
Postage	172.03	
Exchange	.84	
		8,971.50
Profit from Publication		$3,248.36

GENERAL PROFIT AND LOSS STATEMENT.

This statement shows the result of the complete transactions of the Society for the year.

	Loss.	*Profit.*
Profit from Publication		$3,248.36
Interest and dividends		592.91
Current Profit and Loss Account		467.85
		$4,309.12

Banquet, 1922	$189.22	
Membership campaign, 1922	228.96	
Salaries	2,500.00	
General expense	313.60	
		3,231.78
Net profit		$1,077.34

BALANCE STATEMENT.

The statement shows the assets and liabilities on December 31, 1922, and the net assets (capital) on January 1, 1922, and the resultant net profit.

Assets December 31, 1922:

Cash		$3,128.67
Accounts receivable, advertisements		1,293.45
Accounts receivable, dues		942.50
Accounts receivable, sales		195.43
Accounts receivable, subscriptions		215.65
Bonds:		
Army and Navy Club 5%, $1,000		1,000.00
Swift & Co., 5%, $1,000		970.00
Hocking Valley R. R., 4½%, $2,000		1,780.00
N. Y. Steam Corporation, 6%, $1,000		990.00
International Mercantile Marine, 6%, $1,000		940.00
Ohio Power Co., 5%, $1,000		935.00
Pennsylvania Power & Light Co., 5%, $1,000		927.50
Washington Ry. & Electric Co., 4%, $6,000		4,380.00
War Savings Stamps, due 1/1/23		1,000.00
Furniture		104.50
Miscellaneous asset:		
Wm. Cramp & Sons Co		4.34
N. Y. Shipbuilding Corp		4.75
		$18,811.79
Liabilities:		
Dues paid in advance	$134.71	
Subscriptions paid in advance	968.40	
		1,103,11
Net assets December 31, 1922		$17,708.68
Assets January 1, 1922		16,631.34
Net profit		$1,077.34

MEMBERSHIP STATUS.

Honorary:		
Total January 1, 1922		12
Admitted pursuant to new By-Laws adopted July 8, 1922	4	
Transferred from Naval Membership pursuant to new By-Laws adopted July 8, 1922	7	
	11	
Died	1	
Net gain		10
Total December 31, 1922		22
Naval:		
Total January 1, 1922		721
Admitted	128	
Reinstated	9	
	137	
Transferred to Honorary	7	
Resigned	38	
Died	7	
Dropped	10	
	62	
Net gain		75
Total December 31, 1922		796
Civil:		
Admitted		3
Transferred from Associate		13
Total December 31, 1922		16
Associate:		
Total January 1, 1922		452
Admitted	10	
Reinstated	2	
	12	

Transferred to Civil or Naval	17	
Resigned	20	
Died	5	
Dropped	22	
	64	
Net loss		52
Total December 31, 1922		400
Grand total membership, December 31, 1922		1,234
Grand total membership, January 1, 1922		1,185
Grand net gain		49

SUBSCRIBERS.

Total January 1, 1922		575
Stopped	142	
Added	39	
Net loss		103
Total December 31, 1922		472

ELECTION OF OFFICERS.

A meeting of the Council of the Society was held on Thursday, January 4, 1923, for the purpose of receiving the report of the Committee appointed to count ballots cast for officers for 1923. As a result of the vote and pursuant to the By-laws requiring that at least one member of the Council shall be a Naval Constructor, one a Coast Guard Officer and two members shall be civilians, the following were declared elected:

President:

Captain J. T. Tompkins, U. S. Navy.

Secretary-Treasurer:

Commander S. M. Robinson, U. S. Navy.

Council:

Rear Admiral B. C. Bryan, U. S. Navy.

Rear Admiral Robert Stocker, (CC), U. S. Navy.

Captain P. B. Dungan, U. S. Navy.
Captain H. L. Wyman, U. S. Navy.
Engineer-in-Chief Q. B. Newman, U. S. Coast Guard.
Mr. Charles F. Bailey.
Dr. Edgar Buckingham.

ANNUAL DINNER, 1923.

The question of holding a dinner in 1923 having been placed before the membership and decision in the affirmative having been made by a vote of 180 to 35, arrangements have been made to hold a dinner at the New Willard Hotel, Washington, D. C., on Tuesday, February 27, 1923. It is hoped that this dinner will be even more successful than on previous occasions. The committee in charge of the event is as follows:

Commander F. J. Cleary, U. S. Navy.
Commander O. L. Cox, U. S. Navy.
Commander H. S. Howard (CC), U. S. Navy.

The following members have joined the Society since the publication of the last previous JOURNAL:

NAVAL MEMBERS.

Almy, Edmund D., Lieutenant Commander, U. S. N.
Armstrong, Charles A., Lieutenant, U. S. N.
Beltz, Fred S., Lieutenant, U. S. N.
Canoose, John E., Lieutenant, U. S. N.
Carroll, Penn L., Lieut. Commander, U. S. N.
Parsons, Eli B., Lieutenant, U. S. N.
Preston, Charles F., Captain, U. S. N.
Shaw, William Thomas, Lieutenant, U. S. N.
Tinsley, Warwick M., Lieutenant, U. S. N.

CIVIL MEMBER.

Hoffman, W. E., Marine Sales Dept., General Electric Co., Schenectady, N. Y.

ASSOCIATE MEMBERS.

Clausen, H., Engineer Lieutenant, Royal Danish Navy.

Fenlon, Paul B., Representing Worthington Pump and Machinery Corporation, Homer Building, Washington, D. C.

The following have been transferred from Associate Membership to the new Civil Membership:

Herbert, Frederick D., President Kearfott Engineering Company, 95 Liberty St., New York City.

Wood, S. L., Assistant to President, Newport News Shipbuilding & D. D. Co., Newport News, Va.

ADDRESSES.

DO NOT FAIL to keep the Secretary-Treasurer advised as to your ADDRESS at all times, to ensure the prompt delivery of your JOURNAL.

and
D. C.
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